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HEAT TRANSFER AND FLOW FRICTION CHARACTERISTICS OF A PLATE-FIN TYPE CROSS-FLOW HEAT EXCHANGER WITH PERFORATED FINS

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HEAT TRANSFER AND FLOW FRICTION CHARACTERISTICS

OF A PLATE-FIN TYPE CROSS-FLOW HEAT

EXCHANGER WITH PERFORATED FINS

by

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ABSTRACT

Basic heat transfer and flow friction characteristics are presented for two different plate-fin compact heat exchanger surfaces employing the steady state, steam-to-air testing technique. One surface is a plain triangular fin of stainless steel and the other is a triangular fin fabricated from perforated nickel.

The experimental heat transfer characteristics of the perforated nickel fin obtained by the steady state steam-to-air testing technique, described herein, is compared with the results of an identical fin tested by the maximum slope (or transient test) technique.

Both surfaces tested compared favorably with their corresponding analytical solutions; and the comparison of the perforated fin by the two different test techniques was very good.

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NOMENCLATURE

A = exchanger total heat transfer area (i.e., with perforations), sq ft

A_t = exchanger total heat transfer area (i.e., without perforations), sq ft

A_c = exchanger air side minimum free flow area, sq ft

A_f = fin area, sq ft

A_{fr} = air side total frontal area, sq ft

A_w = wall area between air and steam side of exchanger, sq ft

C = coefficient of discharge of a fluid orifice, dimensionless

C_o = initial value of the coefficient of discharge for calculation purposes, dimensionless

c_p = specific heat of air at constant pressure, BTU/(lbm deg F)

d = diameter of fluid metering orifice, in

D = air orifice duct diameter, in

D_d = hydraulic diameter of duct immediately downstream of test core, in

f_d = friction factor in duct immediately downstream of the test core, dimensionless

F = velocity of approach factor for the fluid metering orifice, dimensionless

F_A = thermal expansion factor for the fluid metering orifice, dimensionless

G = exchanger air flow stream mass velocity, $(\dot{m}/A_c)$, (lbm/hr sq ft)

g_c = proportionality factor in Newton's Second Law, $g_c = 32.2$ (lbm ft)/(lbf sec^2)

GR = humidity ratio of air, grains water vapor/lbm dry air

h = unit conductance for thermal convection heat transfer, BTU/(hr sq ft deg F)

h_c = enthalpy of condensate leaving the test core, BTU/lbm

h_s = enthalpy of steam, BTU/lbm

H = humidity ratio of air, lbm water vapor/lbm dry air

k = fluid thermal conductivity, BTU/(hr sq ft deg F/ft)

k_s = core material thermal conductivity, BTU/(hr sq ft deg F/ft)

K_c K_e = contraction loss coefficient for flow at heat exchanger entrance or exit respectively, dimensionless

l = effective fin length (one-half the fin length from wall to wall), in

l_d = length of duct from downstream face of test core to downstream pressure tap, in

l_w = wall thickness between steam and air side of test core, in

L = total fin length flow direction, ft

$\dot{m}$ = air mass flow rate, lbm/hr

P = pressure, in Hg, lbf/sq in, lbf/sq ft

P_b = barometric pressure, in Hg

q = heat transfer rate, BTU/hr

R = gas constant, (ft lbf)/lbm deg R), (53.35 for air)

r_h = hydraulic radius, (A_cL/A_t), ft, ($4r_h$ = hydraulic dia)

s = cross sectional area of fin, sq ft

$\dot{s}$ = mass rate of steam flow, lbm/hr

t = temperature, deg F

T = absolute temperature, deg R, deg K

U = unit overall thermal conductance, BTU/(hr sq ft deg F)

$\dot{w}_c$ = mass rate of condensate from test core, lbm/hr

x = ratio of pressure differential across the orifice to the pressure upstream of the fluid metering orifice, dimensionless

X_m = humidity correction to the density of air, dimensionless

Y = net expansion factor for a square-edged metering orifice, dimensionless

β = ratio of fluid metering orifice diameter to duct diameter, dimensionless

β = compactness, sq ft/cu ft

$\bar{\beta}$ = compactness for perforated material, including effect of area reduction, sq ft/cu ft

Δ = denotes difference

δ = fin thickness, in

η = temperature effectiveness, dimensionless

η_o = total surface temperature effectiveness for air side, dimensionless

η_s = total surface temperature effectiveness for steam side, dimensionless

σ = ratio of free flow area to the frontal area, A_c/A_{fr}

ρ = density, lbm/cu ft

μ = dynamic fluid viscosity, lbm/hr ft

Dimensionless Groupings:

f = Fanning friction factor in test core

j = Colburn heat transfer modulus ($= N_{st} N_{pr}^{2/3}$)

N_R = Reynolds number, $(4r_h G/\mu)$

N_{St} = Stanton number, $(h/G\ c_p)$, a heat transfer modulus

N_{Pr} = Prandtl number, $(\mu\ c_p/k)$

N_{tu} = number of heat transfer units, $(hA/\dot{m}\ c_p)$

N_{Nu} = Nusselt number, $(h\ 4r_h/k)$, $(N_{Nu} = N_R \cdot N_{St} \cdot N_{Pr})$, a heat transfer modulus

Subscripts:

1 = upstream of the core

2 = downstream of the core

A = absolute pressure, lbf/sq ft abs, lbf/sq in abs

a = air side of test core

f = fin

s = steam side of test core

1. Introduction

A steady state steam-to-air heat transfer testing facility [13] for evaluating new and improved surfaces for compact heat exchangers was recently constructed at the U. S. Naval Postgraduate School. This facility, constructed similar to the one at Stanford University, will be used to obtain basic heat transfer and flow friction characteristics for the Bureau of Ships Heat Transfer Project.

The test facility will be fully described, including figures; and the method of analyzing data, as given in complete detail by Ward [13], will be presented in summarized form.

The Harrison heat exchanger was used to continue to check-out the test facility and to make modifications as necessary to minimize uncertainties. An evaluation of the instrumentation and the test results was made to verify that the test results are acceptable.

2. Description of the Test Apparatus

General. This steady state, steam-to-air, heat transfer testing facility [13] is designed to test compact cross-flow heat exchangers. The facility, in its present configuration, is capable of testing heat transfer cores, whose frontal areas, are six inches by six inches. Larger cores with frontal areas up to 12 inches by 12 inches can be tested with minor

modifications in the testing facility. On the air side, the test surfaces can be tested through an approximate Reynolds number range of 500 to 10,000 with a maximum core pressure drop of 2.16 psi.

The surface of the heat exchanger to be tested is placed into the test core section so that the air passes over this surface. Instrumentation is provided to measure the air temperature before and after the test core, and the rate of air flow.

The steam system is to provide a constant temperature heat source. Steam is introduced slightly superheated (five to 10 degrees of superheat) which, in a short distance after entering the test core, becomes saturated steam. The entering steam state can be determined by pressure and temperature measurements alone and can be carefully regulated. The steam system provides for measuring the dry steam rate and condensate rate, which is then used to calculate the heat energy lost by the steam, and compared with the energy gain of the air. A good energy balance gives confidence in the measurements of flow rates and temperatures.

The test facility and a sample test core can be seen pictorially in Figures 1 through 5. A skematic diagram of the air and steam systems is provided in Figures 6 and 7.

Air system. The air system ducting is made of 16-gage, galvanized steel with one-half inch steel flanges. The entrance section is a three-foot square reducing to a one-foot square, each side of which has the curvature of a quarter ellipse. It is located outside of the building to reduce the temperature gradients in the air and is covered with a fine screen mesh to eliminate foreign matter. The next one-foot square section is five feet long and is followed by a three-foot long transition section to a six-inch square. The next section is 12 inches long and is instrumented with four horizontal thermocouple taps, a piezometer ring and two pitot tube taps for conducting vertical and horizontal velocity surveys. Following the test core is another 12-inch long, six-inch square instrument section containing nine thermocouple taps and a piezometer ring. The next two-feet long transition section expands the six-inch square ducting to a circular section with an inner diameter of 13.875 inches. The following circular section is 14 feet long (12.1 diameters) ahead of and four feet long (3.5 diameters) after the air orifice plate. A piezometer ring is located one diameter upstream of the forward edge of a standard ASME square-edged orifice plate and a half diameter downstream. A thermocouple for measuring the air orifice temperature is located two diameters downstream of the orifice plate.

The air is induced through the air system by a 6,000 cfm, two-stage centrifugal compressor driven by a three-phase, 220-volt, 100-hp motor. The compressor discharges to the outside of the building through a 20-inch square duct. The coarse control of the air flow is made with the discharge valve (blast gate) of the compressor and the fine control is made with the double sliding plate valve at the compressor inlet.

Four standard ASME square edged orifice plates were constructed from one-fourth-inch, type 304, stainless steel. The inner diameter, D, of the orifice metering section is 13.875 inches. The diameter, d, of the orifices and required piping in accordance with ASME Power Test Code [12] are as follows:

d (inches)	$\beta = \frac{d}{D}$	Req'd diameters before orifice	Req'd diameters after orifice
10.406	0.75	14.0	3.8
6.244	0.45	8.9	3.0
3.468	0.25	8.2	2.4
2.081	0.15	8.3	2.0

This choice of diameters enables air flow rates from 250 to 22,000 lbm per hour with overlapping ranges.

Downstream of the test section the ducting is insulated with two-inch fiberglass insulation covered by aluminum foil, and the test core is insulated from the air ducting by 1/8-inch Teflon gaskets.

Steam system. Saturated steam is supplied from the school heating system at 65 psig. As the steam enters the test facility (as shown in Figure 7), it goes through a $1\frac{1}{4}$-inch centrifugal separator and a strainer. Next it enters a $1\frac{1}{4}$-inch air operated pilot controlled pressure reducer where it can be reduced from 45 to 15 psig. If necessary to further desuperheat, water can be injected into the steam just before the steam goes into a two-inch centrifugal separator, which is capable of removing 99 percent of all entrained moisture. It then passes through a second $1\frac{1}{4}$-inch air operated, pilot controlled, pressure reducer and into a short transition section, which has stainless steel shavings in the top to give an even flow distribution. The shavings are held in place by a fine mesh stainless steel screen. Immediately proceeding the test core is a straight section instrumented for pressure and temperature measurements. It is in this section that the steam has been reduced to approximately six psig by the second pressure reducer. This final throttling process produces slightly superheated steam (five to 10 degrees superheat). This small amount of superheat is necessary so that the state of the steam can be determined by pressure and temperature measurements alone. The slightly superheated steam, after traveling a short distance upon entering the core, becomes saturated steam and

functions as a constant temperature heat source. A considerable excess of blow steam is passed through the core to prevent a thick film boundary of condensate from forming on the heat transfer surfaces. After the core, the steam and condensate enter another pressure and temperature instrumented, straight section and into another transition section to a two-inch centrifugal separator, where the condensate is separated out. The condensate leaves the system via a floating type steam trap and is subcooled in a small tap water counter-flow heat exchanger to prevent it from flashing into steam when it is collected in a bucket for weighing. The essentially dry steam, called "blow" steam, exiting from the separator, is measured by a standard ASME square edged orifice with flange pressure taps and is then piped to the atmosphere. The "blow" steam orifice temperature is measured seven pipe diameters downstream of the orifice.

The inside pipe diameter, D_s, at the steam orifice is 1.25 inches. There are 14.0 pipe diameters proceeding and 9.0 pipe diameters after the steam orifice. The two orifice plates selected have the following diameters, d_s:

d_s(in)	$\beta_s = \frac{d_s}{D_s}$	Req'd diameters before orifice	Req'd diameters after orifice
0.700	.560	7.8	3.2
0.890	.712	13.0	8.8

The required pipe diameters before and after the orifice is as specified by ASME Power Test Codes [12].

The entire steam system is well insulated to minimize the heat losses from the steam.

The two steam pressure reducers are actuated by two ATMO pressure regulators located on the instrument panel. The compressed air for the ATMO pressure regulators is supplied by a Worthington air compressor that supplies 80-100 psi, which is reduced to 76 psi by a reducer before entering the ATMO pressure regulators. This combination is designed to hold the core steam pressure within a tolerance of $\pm$ 0.1 inch of mercury.

An air line from the air supply of the first pressure reducer was fitted into the steam strainer clean-out plug, so that the steam system can be blown dry upon completion of testing.

Pressure instrumentation. For the air system, pressure measurements are provided for gage pressure upstream of the test core, test core pressure differential, air orifice pressure differential and gage pressure upstream of the orifice. Air pressures are measured with well-type single leg water manometers and inclined draft gages. The steam system pressure instrumentation provides for measuring the steam pressure prior to and after the test core, and the

"blow" steam pressure differential across the steam orifice. Steam pressure measurements are made with well-type single leg mercury manometers.

In the air system, each pressure tap consists of four 1/16-inch holes drilled symmetrically around the duct and connected together by a piezometer ring of soldered 1/4-inch copper tubing. Each piezometer ring is connected to its manometer and draft gage through a brass isolation toggle valve. The core upstream pressure is measured from a 30-inch water manometer. The downstream pressure tap of the test core is located sufficiently downstream, so that full pressure recovery of the air is achieved. The test core pressure differential is measured by a one-inch and a three-inch inclined draft gate and a 60-inch water manometer. A three-inch inclined draft gage and a 30-inch water manometer indicate the air orifice pressure differential. The gage pressure upstream of the air orifice is measured by a 60-inch water manometer. An additional 60-inch water manometer is connected to measure the air orifice pressure differential. This manometer faces the blower and double sliding plate valve, thus allowing continuous visual inspection, while this pressure differential is being adjusted.

On the steam side, the steam pressure above and below the test core and the steam orifice differential are measured

by three 30-inch mercury manometers. All of the steam pressure taps have 1/16-inch holes and insulated 5/8-inch copper tubing leading to four water pots located at the same level above the test core section. This large diameter tubing was selected to permit any condensing steam in these lines to flow back into the steam system. A head of water from three of the water pots, passes through a stainless steel isolation toggle valve, and connects to the mercury wells of the three steam manometers. The fourth water pot has a head of water under it that connects to the top of the steam orifice pressure differential manometer. All of the connecting lines from the water pots are 1/4-inch stainless steel tubing. The condensation that forms in the three water pots maintains a constant water level in the pots and any change in the water level over the mercury, less than 0.01 inches, has a negligible effect on the manometer readings.

Temperature instrumentation. Copper-constantan thermocouples manufactured by Honeywell under the trade name of Megapak are employed to measure all temperatures. The thermocouple measuring junction is at the end of a "sheath" of 1/8-inch stainless steel tubing. The insulated leads extend back to a "head", which houses the terminals for the extension leads to the temperature recorder. There are two types of these thermocouples used. The type used to measure

the air system temperatures are "exposed", meaning that the dissimilar metal junction is extended one sheath diameter beyond the end of the sheath. Those used for the steam system are classified as "remote", since the junction is one sheath diameter short of the end and the end of the sheath is sealed against pressure. All extension wire is polyvinyl covered, 24-gage copper-constantan.

A Honeywell "Electronik 16" Multipoint Strip Chart Recorder senses the thermocouple voltages. An ice bath is used as the reference junction.

In the air system, the air temperature upstream and downstream of the core and the orifice temperature are recorded. The upstream air temperature is obtained by averaging three thermocouples, which are placed in a mid-height horizontal plane and equally spaced across the duct. The downstream temperature is measured by a three-by-three grid-like arrangement of nine thermocouples and the readings averaged. The air orifice temperature is measured by one thermocouple downstream of the orifice. Steam temperatures before and after the core and downstream of the steam orifice are all measured by one thermocouple at each location. Each thermocouple penetration is fitted with a Swagelok compression fitting, so that the insertion length is controllable.

The thermocouples are numbered as shown in Figures 6 and 7 and as indicated next to the face of the recorder.

3. Method of Analyzing Data

General. This section presents the method of analysis of the basic laboratory data to obtain the heat transfer and flow friction characteristics of the surfaces being tested. These equations are essentially those developed by Kays [5] for a similar facility at Stanford with additional amplification by Ward [13] where necessary to permit incorporating into a digital computer program. Equations are presented for computing the air flow rate; heat transfer calculations; Stanton's, Prandtl's, and Reynolds' numbers; Colburn-j; friction factor; and energy balance equations.

Air flow metering. The mass rate of air flow, $\dot{m}$, through an ASME square edged orifice is specified in the ASME Power Test Code [12] as:

$$\dot{m} = 359\ C\ F\ d^2\ F_A\ Y \sqrt{\Delta P_o\ \rho_o}\ \text{lbm/hr}$$

where

C = coefficient of discharge

F = velocity of approach factor

d = orifice diameter, in

F_A = factor for the thermal expansion of primary element

Y = net expansion factor

ΔP_o = pressure differential across the orifice, in H_2O

ρ_o = density of the air upstream of the orifice, lbm/cu ft

The velocity of approach factor, from ASME Power Test Code [10], is:

$$F = \frac{1}{\sqrt{1 - \beta^2}}$$

where

$$\beta = \frac{d}{D}$$

d = orifice diameter, in

D = duct diameter = 13.875 in

The approximate equation for the factor which accounts for the thermal expansion of the type 304 stainless steel primary element was derived by Ward [13] from a plot of the factor in ASME Power Test Code [12]. This factor is:

$$F_A = 1.0 + (t_o - 68)\ (1.85 \times 10^{-5})$$

where t_o = orifice temperature, deg F

The net expansion factor, from ASME Power Test Code [12], is:

$$Y = 1.0 - \left(\frac{x}{1.4}\right)(0.41 + 0.35\ \beta^4)$$

where

$$x = \frac{\Delta P_o\ (0.03605)}{P_{OA}}$$

and

P_{OA} = absolute duct pressure upstream of the orifice, lbf/sq in

$= P_b\ (0.4892) - P_o\ (0.03605)$

P_b = barometric pressure, in H_g

P_o = duct pressure upstream of the orifice, in H_2O (duct pressure is below atmospheric pressure)

The density of the air upstream of the orifice may be found from the equation of state for a perfect gas and modified by a humidity correction factor. Density is:

$$\rho_o = \frac{144\ P_{OA}\ X_m}{53.35\ T_o}$$

where

P_{OA} = absolute duct pressure upstream of the orifice, lbf/sq ft

T_o = absolute air temperature upstream of the orifice, deg R

X_m = humidity correction factor for density as given by Kays and London [6]

$$= \frac{1 + H}{1 + 1.607\ H}$$

H = humidity ratio, lbm water vapor/lbm dry air

The coefficient of discharge is a function of Reynolds number, which in turn is a function of the mass rate of air flow. The determination of C is therefore an iterative process.

Murdock [11] suggested the following equation which is dependent upon the orifice Reynolds number, N_{Ro}:

$$C = C_o + \Delta C \left(\frac{10^4}{N_{Ro}} \right)^{\frac{1}{2}}$$

For various values of β ratio, Murdock [11] gave the following values of the coefficient C_o and ΔC, and a first suggested iteration:

β	C_o	ΔC	C
.15	0.59446	0.00945	0.5975
.25	0.59483	0.01037	0.5966
.45	0.59863	0.01543	0.6014
.75	0.60480	0.05448	0.6128

The orifice Reynolds number is:

$$N_{Ro} = \frac{15.28\ \dot{m}}{\mu_o \quad D}$$

D = duct diameter = 13.875 in

μ_o = dynamic viscosity of air at orifice, lbm/(hr ft)

$$= \frac{0.003527 \quad T_o^{3/2}}{T_o + 110.4} \quad \text{from Hilsenrath [2]}$$

T_o = absolute air orifice temperature, deg K

Heat transfer calculations. For a crossflow heat exchanger, both fluids unmixed, with a constant steam temperature on one side, the number of heat transfer units, N_{tu}, [5] is:

$$N_{tu} = \ln\left(\frac{t_s - t_1}{t_s - t_2}\right)$$

where the temperatures are measured quantities, and

$$N_{tu} = \frac{UA}{\dot{m}\, c_p}$$

The overall thermal conductance, U, is then:

$$U = \frac{\dot{m}\, c_p\, N_{tu}}{A_a}$$

The unit conductance for thermal convection heat transfer on the air side, as given by Kays [5] and Ward [13] is:

$$h_a = \frac{1}{\eta_o}\left[\frac{1}{\frac{1}{U} - A_{ta}\left(\frac{1}{\eta_s A_{ts} h_s} + \frac{\ell_w}{12 A_{wa} k_{sw}}\right)}\right]$$

where

h_a = unit conductance for thermal convection heat transfer to the air, BTU/(hr sq ft deg F)

η_o = total surface temperature effectiveness for the air side, dimensionless

U = unit overall thermal conductance, BTU/(hr sq ft deg F)

A_{ta} = total heat transfer area on the air side (i.e., without perforations), sq ft (use A_a for perforated fins)

η_s = total surface temperature effectiveness on the steam side, dimensionless

A_{ts} = total heat transfer area on steam side, sq ft

h_s = unit conductance for thermal convection heat transfer on the steam side, BTU/(hr sq ft deg F)

l_w = thickness of the wall separating the steam side from the air side, in

A_{wa} = area of the wall on the air side, sq ft

k_{sw} = thermal conductivity of the wall, BTU/(hr sq ft deg F/ft)

On the steam and air sides, when extended surfaces are employed, the overall temperature effectiveness, η , is given by [6] as:

$$\eta = 1 - \frac{A_f}{A}(1 - \eta_f)$$

where

A_f = fin transfer area, sq ft

A = total transfer area on one side, sq ft

η_f = fin temperature effectiveness, dimensionless

A good approximation of fin temperature effectiveness for most extended fin geometrics [6] is:

$$\eta_f = \frac{\tanh\ (ml)}{ml}$$

where

$$m = \sqrt{\frac{24h}{k_{sf}\ \delta}}$$

and

k_{sf} = thermal conductivity of the fin, BTU/(hr sq ft deg F/ft)

δ = fin thickness, in

On the steam side, the value of the thermal convection heat transfer, h_s, is assumed to be 2,000 BTU/(hr sq ft deg F) [5]. A $\pm$100 percent difference in actual value introduces only an error of $\pm$.6 percent for a Reynolds number of 1,000 and a $\pm$2.5 percent error for a Reynolds number of 10,000. On the air side, to determine the thermal convection heat transfer, h_a, to be used to calculate the fin temperature effectiveness, a value of $h_a = U$ is a first approximation. Then by an iterative process, the air side h and η are determined.

<u>Dimensionless groupings (air side)</u>. The Reynolds number, N_R, is evaluated using the hydraulic diameters:

$$N_R = \frac{4 r_h G}{\mu}$$

where the hydraulic radius is defined as:

$$r_h = A_c L / A_t$$

and

μ = fluid viscosity evaluated at the average bulk temperature, lbm/hr ft. Evaluation by temperature dependency equations are given by Hilsenrath [2] and Ward [13]

G = exchanger air flow stream mass velocity, $(\dot{m}/A_c)$, lbm/(hr sq ft)

The Stanton number, N_{St}, is:

$$N_{St} = \frac{h_a}{G\ c_p}$$

The Prandtl number, N_{Pr}, is:

$$N_{Pr} = \frac{\mu\ c_p}{k}$$

where

k = fluid thermal conductivity evaluated at the average bulk temperature in the core, BTU/(hr sq ft deg F/ft)

μ = fluid viscosity evaluated at the average bulk temperature, lbm/hr ft

The Colburn j-factor, the generalized heat transfer grouping is:

$$j = N_{St}\ N_{Pr}^{2/3}$$

Friction factor calculations. The derivation of the core friction factor on the air side has been clearly developed by Kays [5] and Ward [13] and will only be repeated in its final form here:

$$f = \frac{r_h\ \rho_m}{L} \left\{ \frac{\Delta P_c (4.3255 \times 10^9)}{G^2} - \left[\frac{K_c - (1 + \sigma^2)}{\rho_1} \right] - \left[\frac{K_e + 1 + \sigma^2 \left(1 + 4\ \frac{f_d\ l_d}{D_d} \right)}{\rho_2} \right] \right\}$$

where

r_h = hydraulic radius, $(A_c L/A_t)$, ft, ($4r_h$ = hydraulic dia)

ρ_m = mean density in the core, lbm/cu ft

$$= \frac{T_1 \rho_1 + T_2 \rho_2}{2} \left[\frac{1}{T_s - \left(\frac{T_1 - T_2}{N_{tu}} \right)} \right]$$

L = total exchanger flow length, ft

ΔP_c = pressure drop across the core, in H_2O

G = exchanger air flow stream mass velocity, lbm/(hr sq ft)

K_c, K_e = contraction loss coefficient for flow at heat exchanger entrance or exit respectively, dimensionless (these valvues are calculated by Ward [13] as a function of free flow/frontal area and Reynolds number)

σ = ratio of free-flow area to frontal area

ρ_1, ρ_2 = density upstream or downstream of the core, respectively, lbm/cu ft

f_d = friction factor for the duct downstream of the core, dimensionless

l_d = length of duct from the core to the downstream pressure tap = 11.12 in

D_d = hydraulic diameter of the duct downstream of the core = 6.0 in

The duct friction factor, f_d, is assumed to be a constant of .0051, corresponding to a nearly smooth pipe for a Reynolds number range of 100,000 to 500,000 (McAdams [10], page 156).

Energy balance calculations. As given by Ward [13], the energy gain of the air, q_{air}, is:

$$q_{air} = \dot{m}\, c_p\, (t_2 - t_1)\ , \text{BTU/hr}$$

The total energy loss of the steam, q_{steam}, is that given up by the excess blow steam and that given up by the condensate in the core:

$$q_{steam} = \dot{s}\, (h_{s1} - h_{s2}) + \dot{w}_c\, (h_{s1} - h_c)\ , \text{BTU/hr}$$

where

$\dot{s}$ = mass flow rate of excess steam, lbm/hr

$\dot{w}_c$ = mass flow rate of condensate, lbm/hr

h_{s1} = inlet steam enthalpy, BTU/lbm, evaluated from pressure and temperature measurements at core inlet

h_{s2} = enthalpy of saturated steam evaluated at the core downstream pressure, BTU/lb

h_c = enthalpy of saturated liquid evaluated at the core downstream pressure, BTU/lb

Ward [13] clearly specifies the method for determining all the equation elements. Briefly, the mass rate of excess steam is determined similarly to the mass rate of air flow,

the condensate rate being the condensate collected over the period of the run, and the enthalpies are close approximations from plotting values in Keenan [7].

The error in the energy balance is:

$$\text{ERROR} = \frac{q_{air} - q_{steam}}{q_{air}} \times 100$$

4. Evaluation of Instrumentation

Temperature check. A check of the temperature recording system was made with seven thermocouples (numbers 2,3,8,9,11 and 14) and a calculated mercury thermometer. The thermocouples and thermometer were suspended vertically into an insulated coffee can full of water, with a heater element and a mixer. A temperature correlation, after a calibration check of the multipoint recorder, was made at room temperature. After the mixer had been on for several minutes and equilibrium had been reached in the can, the thermocouple readings in millivolts (mv) and thermometer readings in deg F were recorded and compared.

The average of the thermocouple readings were within .31 deg F and the scatter in the readings was $\pm$.45 deg F. For a tabulation of the results, see Table I, Appendix IV.

Steam saturation state check. When the steam exits from the core, it is in the mixed phase region and its temperature and pressure are interrelated. Since the steam temperature and pressure are measured at this point, two randomly selected

runs were checked and their accuracy evaluated. A check was made from a run using the Harrison core, and one using the Solar core. Both runs had the same results. Using the steam absolute pressure, the corresponding saturation temperature from Keenan and Keyes [7] was 0.4 deg F lower than the measured temperature. See Table II, Appendix IV for the calculations.

Hot and cold core friction data comparison. For a hot core test, both heat transfer and flow friction data are determined. Flow friction data only is taken during a cold core test. A comparison of flow friction data was compared for the solar core and data points fall almost on top of each other. Since the small differences in some cases are both above and below the curve, this difference is attributed to experimental scatter.

Overlapping flow rates. The air orifice plates were chosen such that the ranges of each orifice plate would overlap with the next larger or smaller, so comparison of data could be accomplished. With the Harrison heat exchanger two runs were made; one run was with the smallest air orifice plate (β = .15) and the other run was with the next larger air orifice plate (β = .25). The results from this test are given below:

Run	β	ΔP_o	$\dot{m}$	T_1	T_2	ΔP_{core} (in Hg)
2	.15	20.9	1071.8	53.5	159.5	2.34
4	.25	2.56	1056.9	51.1	158.9	2.37

Run (cont'd)	j	f	N_R
2	.00270	.00963	1864
4	.00268	.01022	1846

The Colburn j-factors are very close, within .7 percent and the flow friction values are within 5.8 percent. This friction value difference was investigated and is attributed to several factors. The runs weren't exactly duplicated, the difference in mass flow rate, which is squared in determining f, accounts for 1.7 percent; the core differential pressure accounts for 1.3 percent; and the rest of the difference is attributed to experimental scatter and inherent inability for complete reproducibility in the test facility.

Energy balance. An energy balance comparing the heat transferred from the steam with the heat transferred to the air, provides a check on the temperature and the flow measurements.

5. Presentation of Test Results

Description of tables and graphs. All of the core dimensions necessary to reduce the basic laboratory test data are summarized under TEST CORE GEOMETRICAL DATA, Table I of Appendix I. The necessary dimensional data (f_d, l_d, and D_d) of the test facility is given in the Method of Analyzing Data.

The reduced laboratory test results are shown in Table II of Appendix I. These results contain the heat transfer

and flow friction characteristics from the hot core tests, and isothermal friction data from the cold core tests.

The test results are presented in both tabular and graphical form. The heat transfer and flow friction characteristics for the air-side surface are presented using the Colburn dimensionless heat transfer modulus, $j = N_{St} N_{Pr}^{2/3}$ versus N_R and f, the dimensionless Fanning friction factor, versus N_R. Table III of Appendix I is the tabular form of the results; and Figures 8 and 9, the graphical form.

In addition to the separate pair of curves for each surface geometry, a summary and comparison curve is also presented in Figure 11. Included on the figure are the analytical solutions for an equilateral triangle and a rectangular configuration [9].

The heat transfer characteristics of the Solar No. 2 perforated nickel fin surface obtained by the steady state steam to air testing technique is compared with the results of an identical fin surface tested by the maximum slope technique and presented in Figure 11.

Description of surfaces tested. Two surfaces of the triangular fin configuration were tested and their general characteristics are shown below:

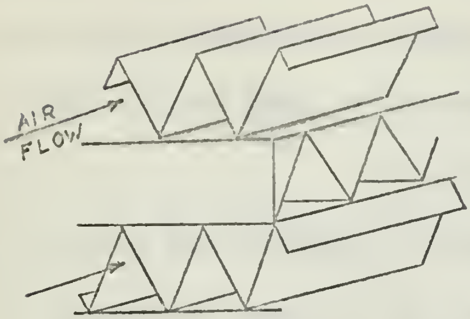

Diagram A
Plain Triangular Fin

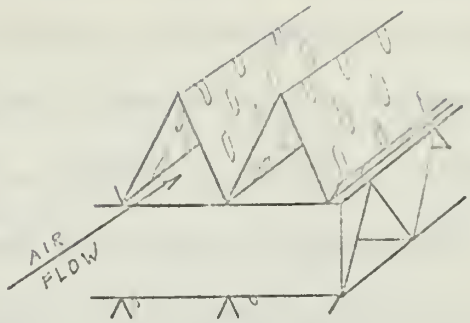

Diagram B
Perforated Triangular Fin

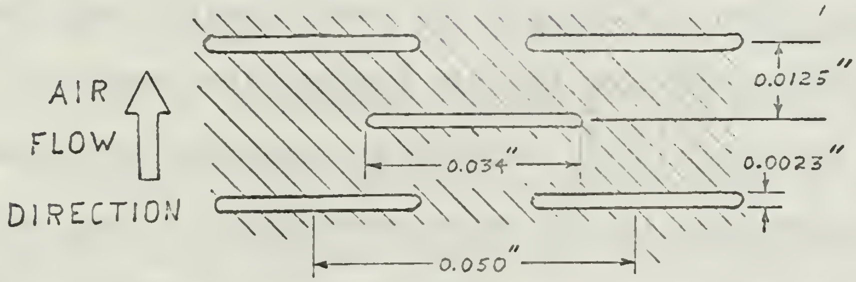

Diagram C
Perforated Fin Material, type 80/20T

The first surface tested is Harrison Surface No. 1. This is one of two identical surfaces in the Harrison heat exchanger. It is a solid, plain triangular surface of stainless steel. The Solar heat exchanger has two different surfaces. Solar Surface No. 1 is a solid nickel, plain triangular fin surface; Surface No. 2 is a triangular fin surface of perforated nickel. This perforated fin, type 80/20T,

(Reference: Perforated Products, Inc., Catalog No. 9-4, page 6) is shown above. Of the two Solar surfaces, only Surface No. 2 was tested. The geometrical data on all the surfaces is given in Table I, Appendix I.

The tabular Summary of Basic Heat Transfer and Flow-Friction Data, Table III of Appendix I, is taken directly from the curves representing the best interpretation of the calculated test points.

6. Discussion of Results

From the summary curve, Figure 10, for Reynolds numbers less than 1,000, both the heat transfer and flow friction characteristics exhibit laminar flow behavior. In Briggs and London [1], analytical solutions are presented for fully developed laminar flow convection in long cylinders of triangular cross-section. The expressions developed for an equilateral triangle are:

$$f = \frac{13.33}{N_R} \tag{1}$$

$$N_{St} \cdot N_R \cdot N_{Pr} \ (= N_{Nu}) = 2.35 \tag{2a}$$

Substituting, $N_{Pr} = 0.70$ and rearranging terms, equation (2a) becomes:

$$N_{St} \cdot N_{Pr}^{2/3} = \frac{2.65}{N_R} \tag{2b}$$

London [9] obtained an analytical solution for laminar flow for a plain rectangular fin where the cross sections have

one dimension four times as great as the other. The resulting expressions are:

$$f = \frac{18.3}{N_R} \tag{3}$$

$$N_{St} \cdot N_R \cdot N_{Pr} \ (= N_{Nu}) = 4.65 \tag{4a}$$

and for $N_{Pr} = 0.70$, equation (4a) becomes:

$$N_{St} \cdot N_{Pr}^{2/3} = \frac{5.25}{N_R}$$

The lines representing equations (1), (2b), (3), and (4b) have been drawn on Figure 11.

The experimental heat transfer and flow friction curves for Harrison Surface No. 1 are close to the analytical solutions for an equilateral triangle. The Solar Surface No. 2 is compared with the curves for the four by one rectangular cross-section, analytical solutions and lie slightly below them.

An enlarged view of the actual triangular fin geometry of the two surfaces tested is shown in Figure 12 to offer some comparison between the actual surfaces and the analytical solutions. From Figure 12, it can be seen that the Harrison Surface No. 1 fin is almost equilateral in shape and the actual shape of the Solar Surface No. 2 approaches a rectangular cross-section. Therefore, this good correlation in the comparison of the actual and analytical solutions increases the confidence in the data.

As another check on the experimental results from the steady state steam-to-air test facility, a comparison of the heat transfer modulus, Colburn j-factor, obtained from the Solar side No. 2 is made with the data from an identical fin surface employing transient testing techniques. This comparison is shown in Figure 11 and the range of overlapping data is indicated. The greatest accuracy in the maximum slope technique is achieved from an approximate Reynolds number range of 100 to 500. It is for this reason that the slope of the Colburn j-factor curve is drawn through the points from the maximum slope technique in the Reynolds number range of 100 to 500. The results of this comparison are very good.

In Kays [5], an investigation is made into the effects of the value of N_{tu} on accuracy of being able to determine the heat transfer and flow friction characteristics. It is shown that for the consideration of the heat transfer characteristics alone, it is more desirable to operate in the N_{tu} range of 0.5 to 2.0. Considering flow friction behavior only, a slightly higher value is needed. For both heat transfer and flow friction characteristics, the most desirable N_{tu} range is 1.00 to 3.00. One relationship for N_{tu}, based on a two fluid unmixed heat exchanger with a constant steam temperature, is:

$$\frac{(t_s - t_1)}{(t_s - t_2)} = e^{N_{tu}}$$

where

t_s = steam temperature, deg F

t_1 = air inlet temperature, deg F

t_2 = air outlet temperature, deg F

N_{tu} = number of heat transfer units, $(hA/\dot{m}\ c_p)$

It can be seen that, as t_2 approaches t_s, the N_{tu} value will exceed the maximum desired range of 3.00. It is at the very low flow rates that large N_{tu} values are possible, and it is for the N_{tu} consideration that the lower limit of testing was established.

The upper limit was established by Kays [5], by the requirement that the excess "blow" steam rate be at least five times the condensate rate. However, by investigating into the effects on h_s, the unit conductance for thermal convection heat transfer on the steam side, for smaller steam-to-condensate ratios, this requirement may be modified.

Since the heat transfer and flow friction characteristics are dimensionless, the test curves are applicable to flow passages of different dimensions than those tested as long as complete dimensional similarity is maintained.

7. Uncertainty Analysis

The basic heat transfer and flow friction characteristics versus Reynolds number of test surfaces are determined by this test facility. An investigation into the accuracy of

these results will be shown by the method described by Kline and McClintock [8]. The possible sources of error and their respective uncertainty for j, f, and N_R will be derived.

The possible sources of errors emanate from the uncertainties in the:

(1) Physical constants: These values were obtained from references [2] and [3], and their estimated uncertainty is:

$$c_p = \pm .5\%$$

$$k_s = \pm 1.0\%$$

$$N_{Pr} = \pm 2.0\%$$

$$\mu = \pm 1.0\%$$

(2) Geometrical measurements: The probable uncertainties arise from core fabrication errors and linear dimension errors, and their estimated accuracies are:

$$A,\ A_t,\ A_c,\ A_f,\ A_{fr} = \pm 1.0\%$$

$$L = \pm .5\%$$

(3) Instrumentation: The source of these errors is in the temperature and pressure readings.

From the downstream air temperature readings, there is a spread of several degrees, causing an estimated temperature uncertainty of:

$$t = \pm 1.0 \text{ deg F}$$

From the pressure instrumentation, the manufacturer's calibration of manometers and draft gages are assumed to be sufficiently accurate. The possible errors are assumed to be the fluctuations in pressure and are:

$\Delta P_c = \pm\ 1.0\%$

$P_o = \pm\ 1.0\%$

$\Delta P_o = \pm\ 1.0\%$

$P_b = \pm\ .001$ in Hg (negligible)

The method of determining the overall uncertainty, as based upon 20:1 odds [8], is:

$$w_R = \left[\left(\frac{\partial R}{\partial v_1} w_1\right)^2 + \left(\frac{\partial R}{\partial v_2} w_2\right)^2 + \cdots\cdots + \left(\frac{\partial R}{\partial v_n} w_n\right)^2\right]$$

where

w_R = the uncertainty interval in the result

R = the expression for the result

w_1, w_2, w_n = the uncertainty interval of the terms that comprise the result

v_1, v_2, v_n = the terms that make up the results

To determine the uncertainty in N_{tu}, the restriction that N_{tu} be held in the range of 1.0 to 3.0 will be made. The average steam temperature, t_s = 230 deg F, and an average air inlet temperature, t_1 = 70 deg F, will be assumed. The maximum allowable t_2 is then 222 deg F.

$$e^{N_{tu}} = \frac{(t_s - t_1)}{(t_s - t_2)}$$

$$N_{tu} = \ln (t_s - t_1) - \ln (t_s - t_2)$$

$$w_{N_{tu}} = \left[\left(\frac{\partial N_{tu}}{\partial t_s} w_{t_s} \right)^2 + \left(\frac{\partial N_{tu}}{\partial t_1} w_{t_1} \right)^2 + \left(\frac{\partial N_{tu}}{\partial t_2} w_{t_2} \right)^2 \right]^{1/2}$$

$$\frac{\partial N_{tu}}{\partial t_s} = \frac{1}{(t_s - t_1)} - \frac{1}{(t_s - t_2)} = \frac{-(t_2 - t_1)}{(t_s - t_1)(t_s - t_2)} = -.12$$

$$\frac{\partial N_{tu}}{\partial t_1} = \frac{-1}{(t_s - t_1)} = -.00625$$

$$\frac{\partial N_{tu}}{\partial t_2} = \frac{-1}{(t_s - t_2)} = -.125$$

The following are the uncertainty intervals in the terms that comprise the result:

$$w_{t_s} = \pm .4 \, , \; w_{t_1} = \pm .4 \, , \; w_{t_2} = \pm 1.0$$

Substituting in the above values and normalizing,

$$\frac{w_{N_{tu}}}{N_{tu}} = \left\{ \left[(-.12)\left(\frac{.4}{3}\right) \right]^2 + \left[(-.00625)\left(\frac{.4}{3}\right) \right]^2 + \left[(-.125)\left(\frac{1.0}{3}\right) \right]^2 \right\}^{1/2}$$

$$= .045 = \pm 4.5\%$$

An estimated uncertainty in air flow metering is presented by Kays [5]. For this test facility it is estimated to be:

$$\dot{m} = \pm 1.0\%$$

The overall unit thermal conductance, U, is found by:

$$U = \frac{\dot{m} \, c_p \, N_{tu}}{A_a}$$

The overall uncertainty in U is:

$$w_U = \left[\left(\frac{\partial U}{\partial \dot{m}}\, w_{\dot{m}}\right)^2 + \left(\frac{\partial U}{\partial c_p}\, w_{c_p}\right)^2 + \left(\frac{\partial U}{\partial N_{tu}}\, w_{N_{tu}}\right)^2 + \left(\frac{\partial U}{\partial A_a}\, w_{A_a}\right)^2\right]^{1/2} =$$

$$= \left[\left(\frac{c_p N_{tu}}{A_a}\, w_{\dot{m}}\right)^2 + \left(\frac{\dot{m} N_{tu}}{A_a}\, w_{c_p}\right)^2 + \left(\frac{\dot{m} c_p}{A_a}\, w_{N_{tu}}\right)^2 + \left(\frac{\dot{m} c_p N_{tu}}{A_a^2}\, w_{A_a}\right)^2\right]^{1/2}$$

Normalizing,

$$\frac{w_U}{U} = \left[\left(\frac{w_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{w_{c_p}}{c_p}\right)^2 + \left(\frac{w_{N_{tu}}}{N_{tu}}\right)^2 + \left(\frac{w_{A_a}}{A_a}\right)^2\right]^{1/2} =$$

$$= \left[(.01)^2 + (.005)^2 + (.045)^2 + (.01)^2\right]^{1/2} =$$

$$= .047 = \pm\ 4.7\%$$

Next is the determination of the uncertainty in the thermal convection heat transfer on the air side, h_a:

$$h_a = \frac{1}{\eta_o}\left[\frac{1}{\dfrac{1}{U} - \dfrac{A_a}{\eta_s A_s h_s} - \dfrac{A_a\, l_w}{12\ A_{wa}\, k_{sw}}}\right]$$

$\eta_o = \pm\ 5.0\%$

$\eta_s = \pm\ 5.0\%$

$h_s = \pm\ 2.5\%$

$k_{sw} = \pm\ 2.0\%$

$l_w = \pm\ 1.0\%$

All A_x's $= \pm\ 1.0\%$

It is estimated that the uncertainty in the thermal convection heat transfer on the air side is:

$$h_a = \pm\ 5.0\%$$

By a similar process, the uncertainties in the other calculated quantities are:

$$j = N_{st}\ N_{Pr}^{2/3} = \pm\ 6.0\%$$

$$f = \pm\ 6.0\%$$

$$N_R = \pm\ 2.0\%$$

8. Conclusions and Acknowledgements

From the steady state, steam-to-air, compact heat exchanger testing facility, the basic heat transfer and flow friction characteristics have been presented for Harrison Surface No. 1 and Solar Surface No. 2. The characteristics of these two surfaces compared very favorably with analytical solutions for correspondingly similar fin configurations.

The experimental heat transfer characteristics of Solar Surface No. 2 obtained by the steady state, steam-to-air testing technique compared favorably with the results from an identical surface employing the transient test (or maximum slope) technique.

From the checks on the accuracy of the instrumentation, as described in Evaluation of Instrumentation, and the results of the testing, as summarized above, the uncertainties in the testing facility have been sufficiently reduced to the

extent that the accuracy of the basic heat transfer and flow friction characteristics are acceptable.

The author expresses sincere appreciation to Dr. Paul F. Pucci, Professor of Mechanical Engineering, for his patience, assistance, and encouragement. He also extends his gratitude to Mr. Joe Beck for his technical assistance in performing the necessary modifications to the testing facility. The U. S. Naval Bureau of Ships is also thanked for providing the necessary financial support.

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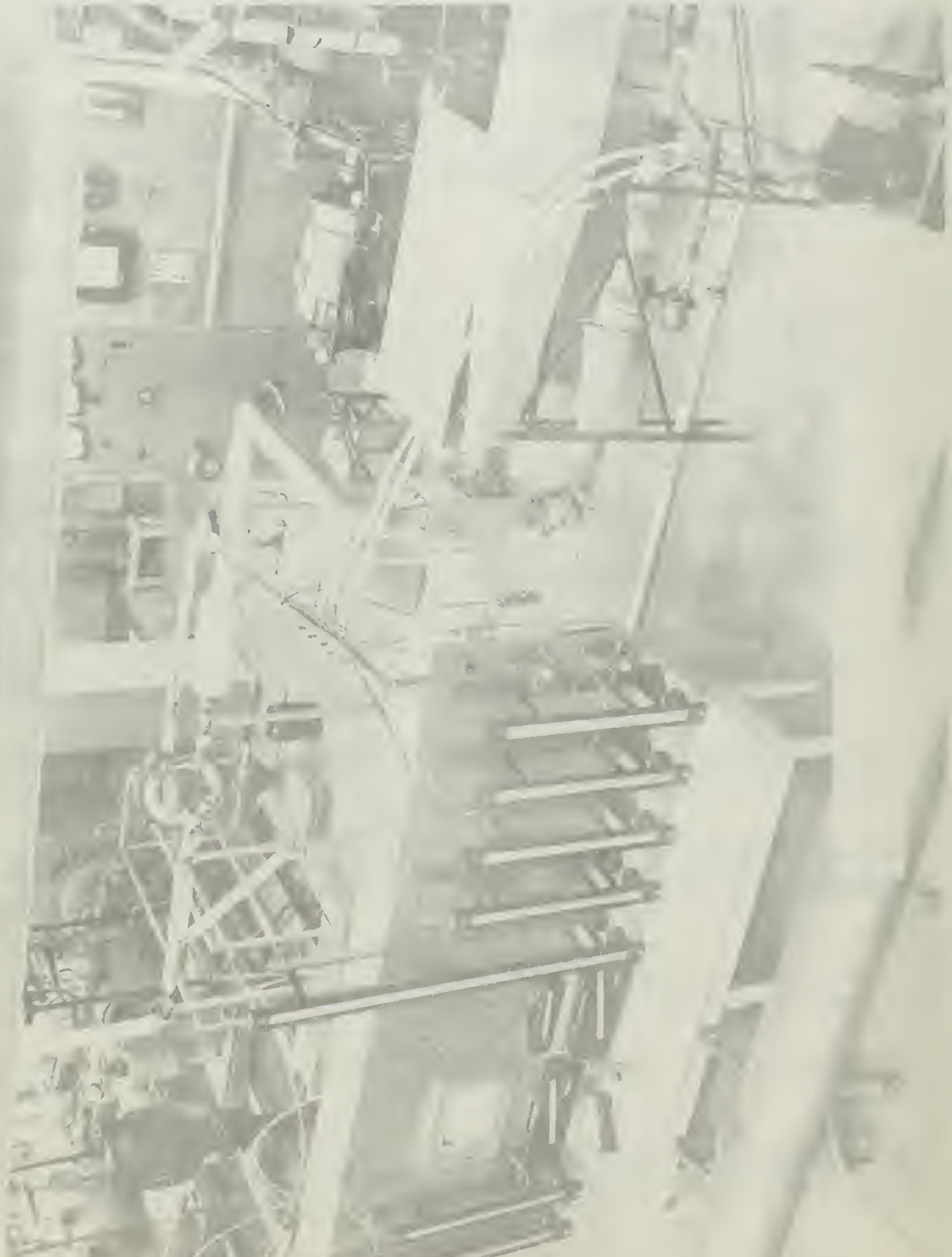
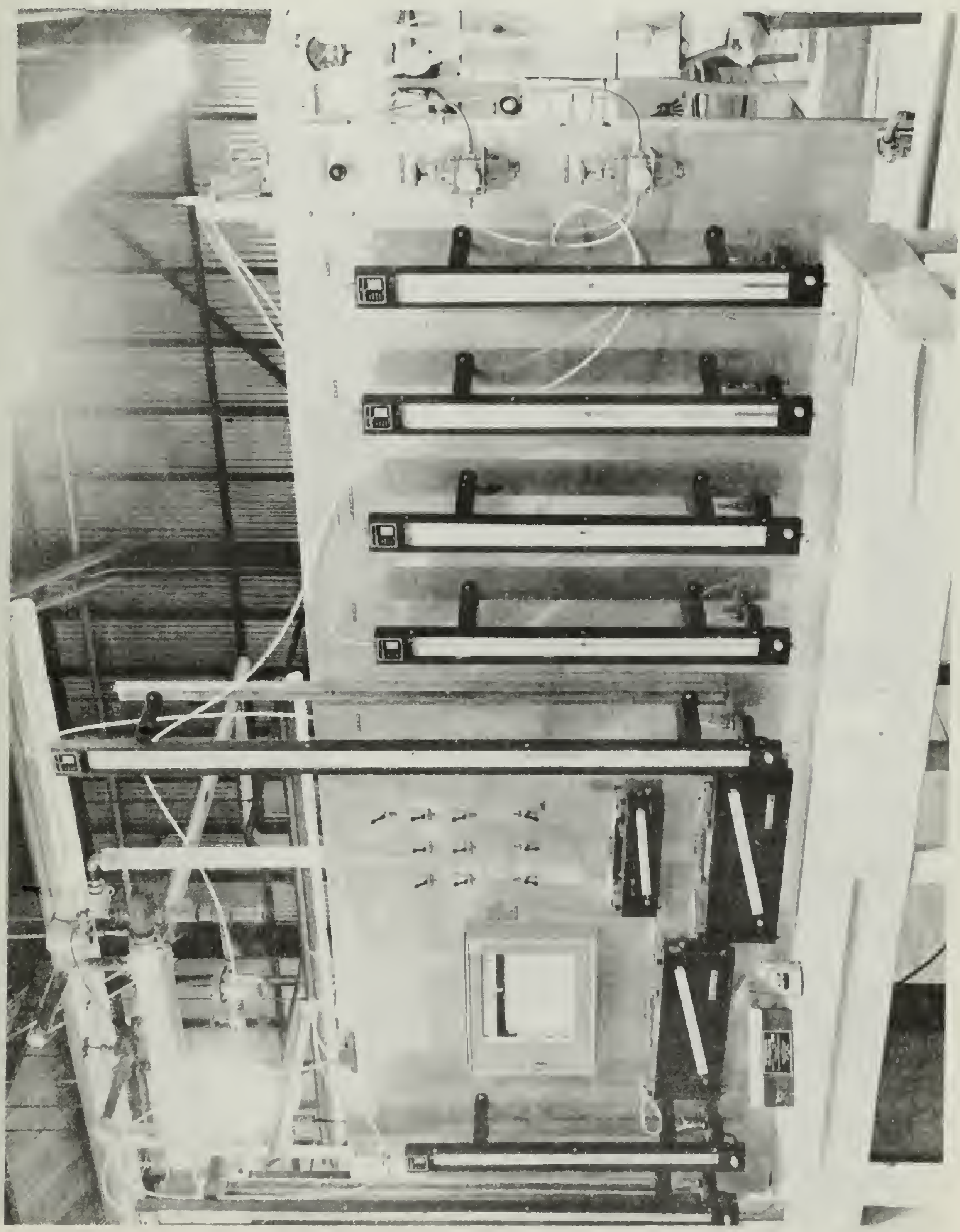
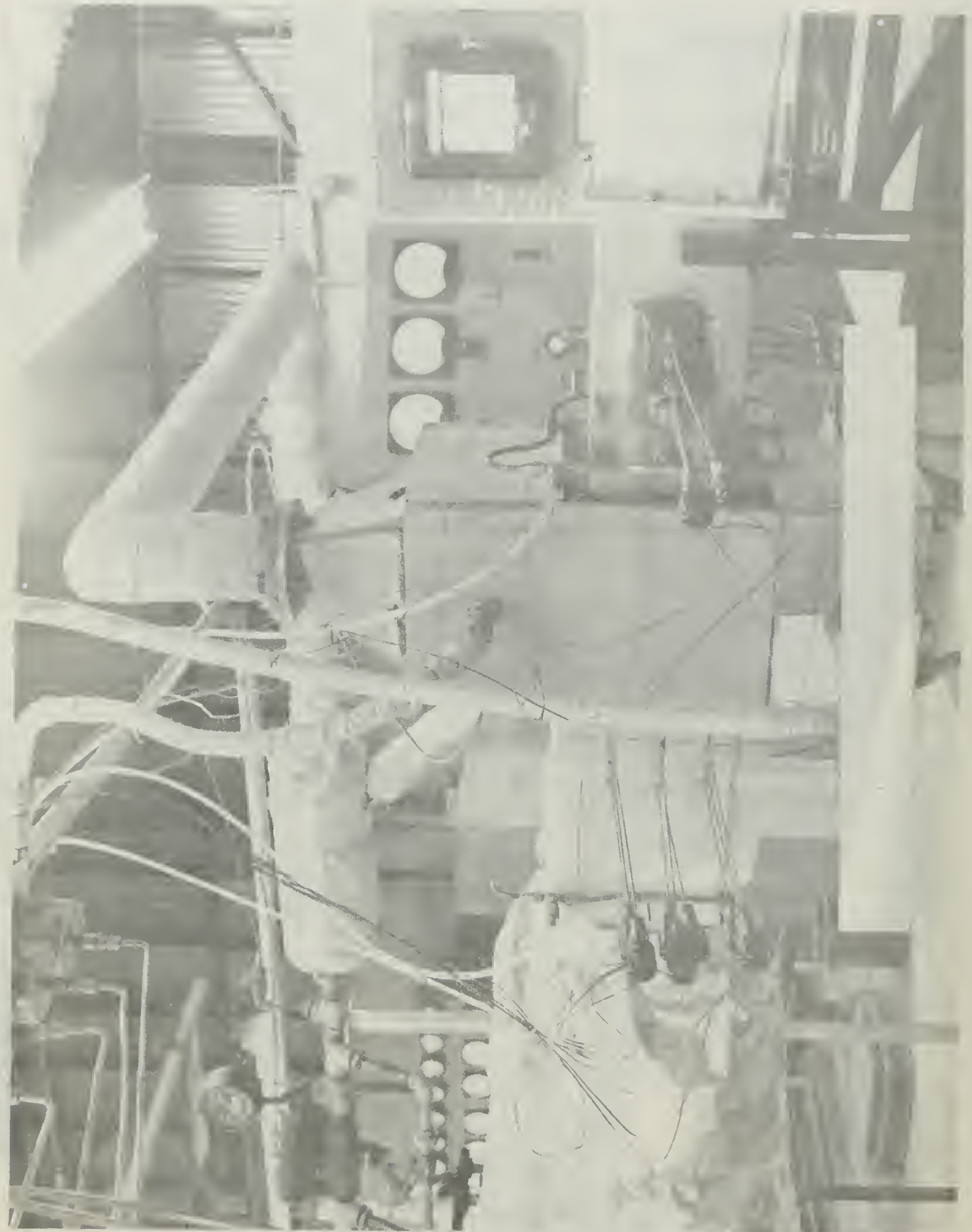
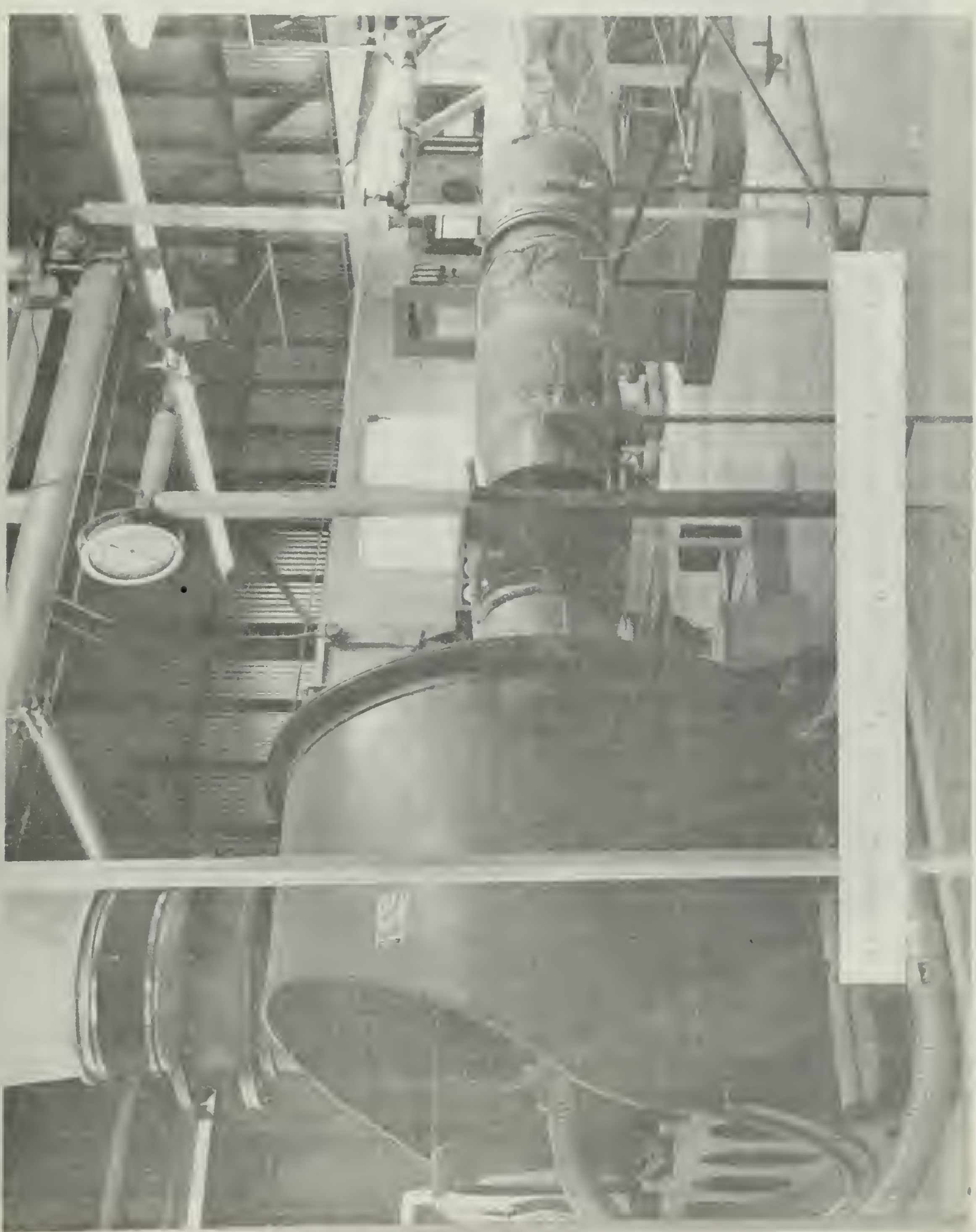
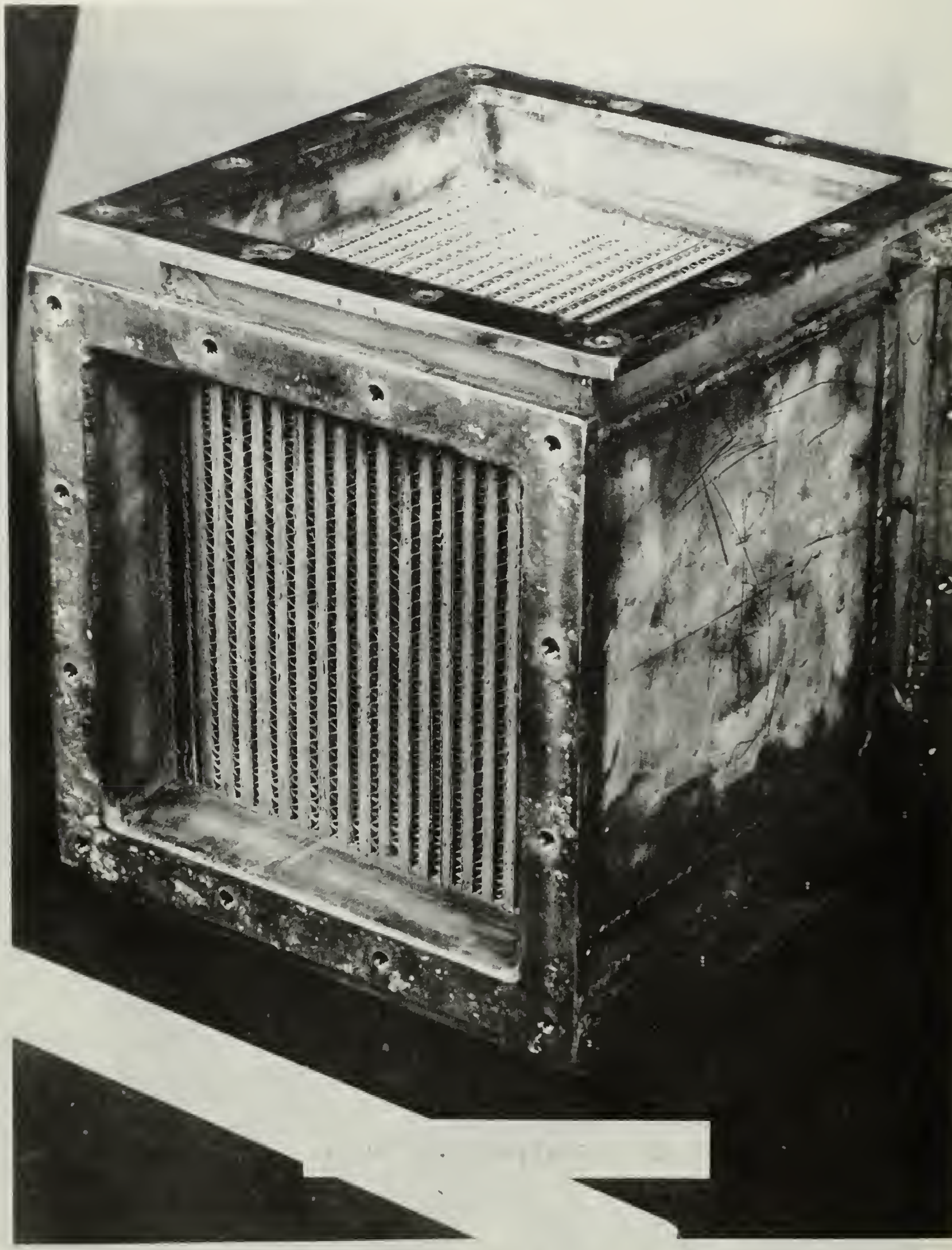
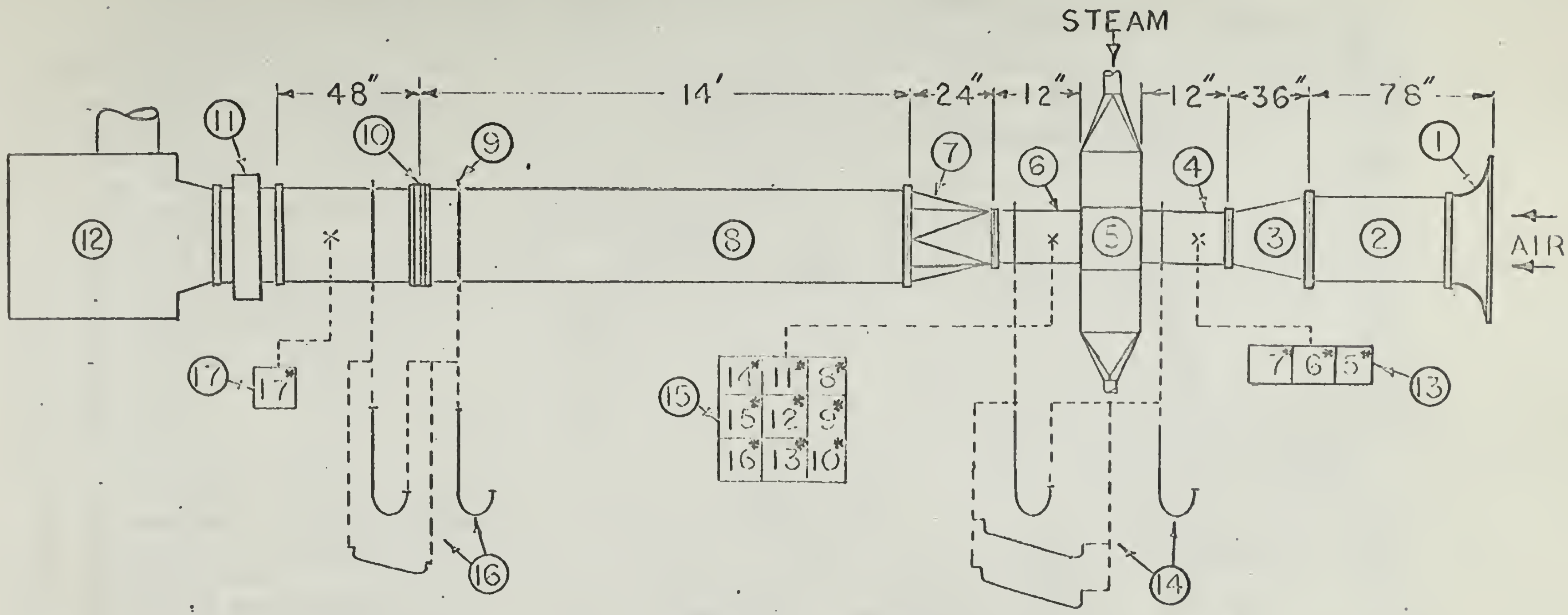








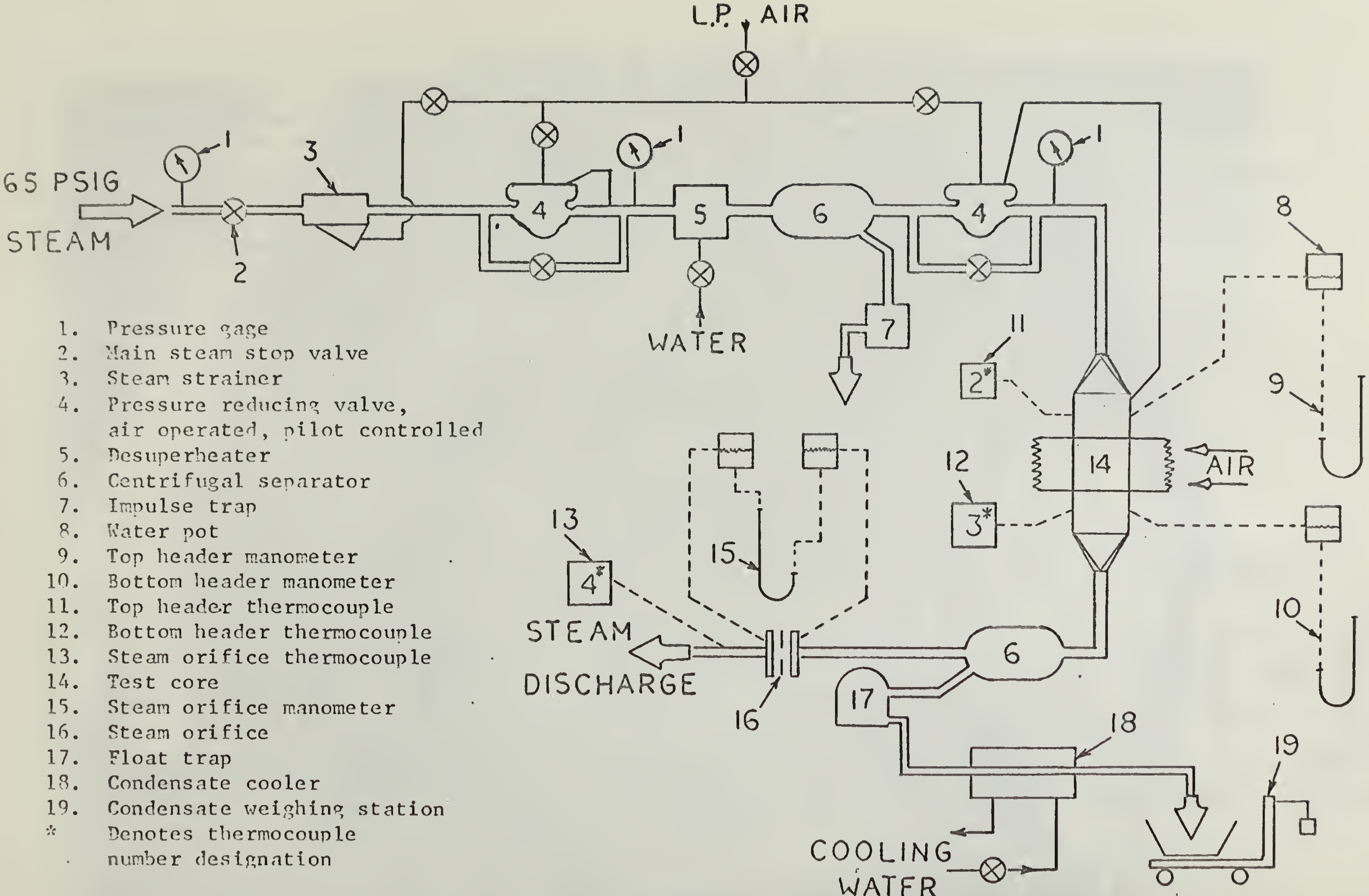




1. Entrance section
2. Upstream ducting, 12" x 12"
3. Transition to test core dimensions
4. Upstream instrumentation section
5. Test core
6. Downstream instrumentation section
7. Transition to 14" diameter duct
8. Downstream ducting, 14" diameter
9. Piezometer ring
10. Air metering orifice
11. Double sliding plate valve
12. Turbo compressor
13. Upstream thermocouples
14. Test core manometers and draft gages
15. Downstream thermocouples
16. Air orifice manometers and draft gages
17. Air orifice thermocouple

* Denotes thermocouple number designation

FIGURE 6 AIR SYSTEM



1. Pressure gage
2. Main steam stop valve
3. Steam strainer
4. Pressure reducing valve, air operated, pilot controlled
5. Desuperheater
6. Centrifugal separator
7. Impulse trap
8. Water pot
9. Top header manometer
10. Bottom header manometer
11. Top header thermocouple
12. Bottom header thermocouple
13. Steam orifice thermocouple
14. Test core
15. Steam orifice manometer
16. Steam orifice
17. Float trap
18. Condensate cooler
19. Condensate weighing station

* Denotes thermocouple number designation

STEAM SYSTEM

FIGURE 7

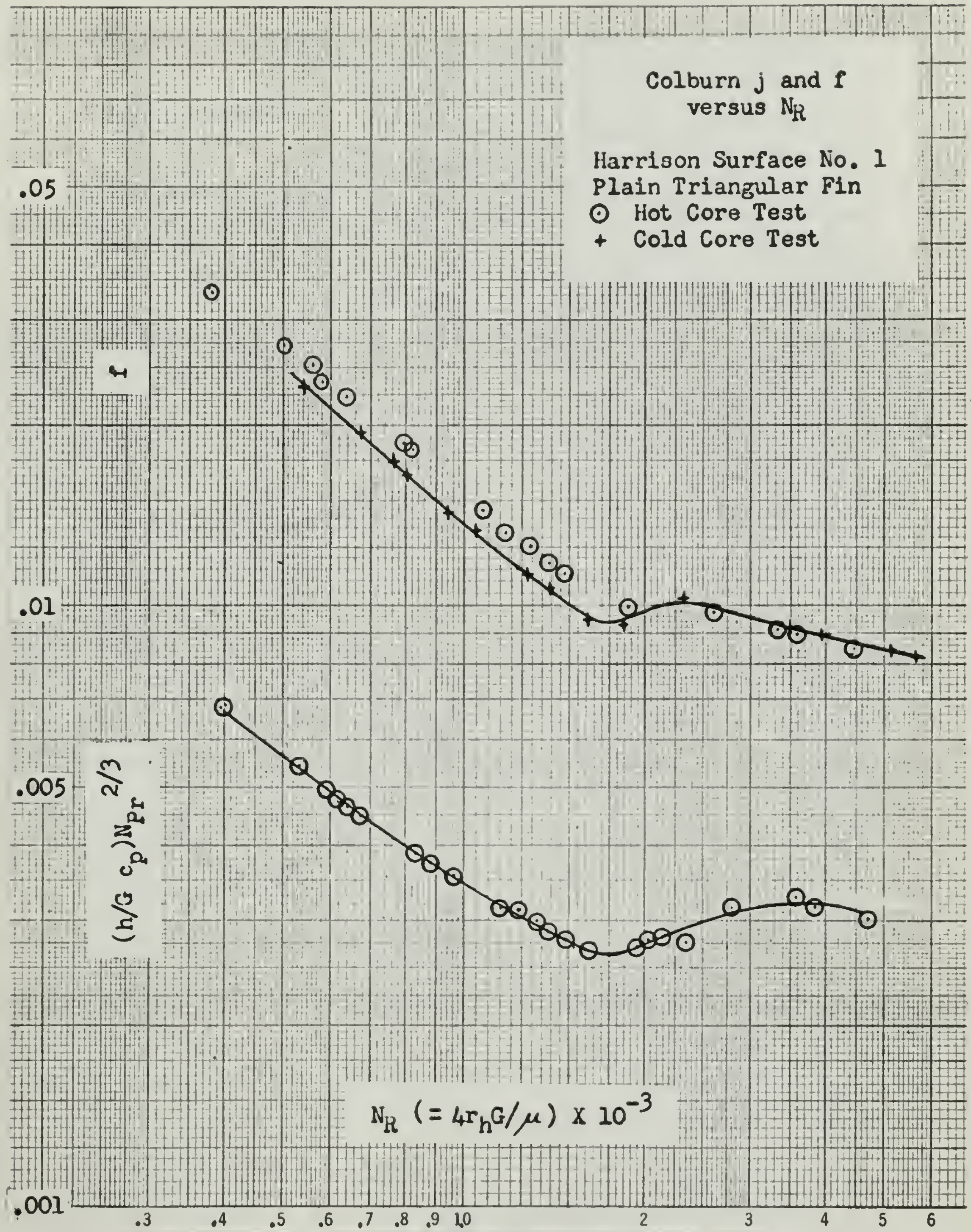


Figure 8. Plate-Fin Surface Heat Transfer and Friction Data, Harrison Surface No. 1

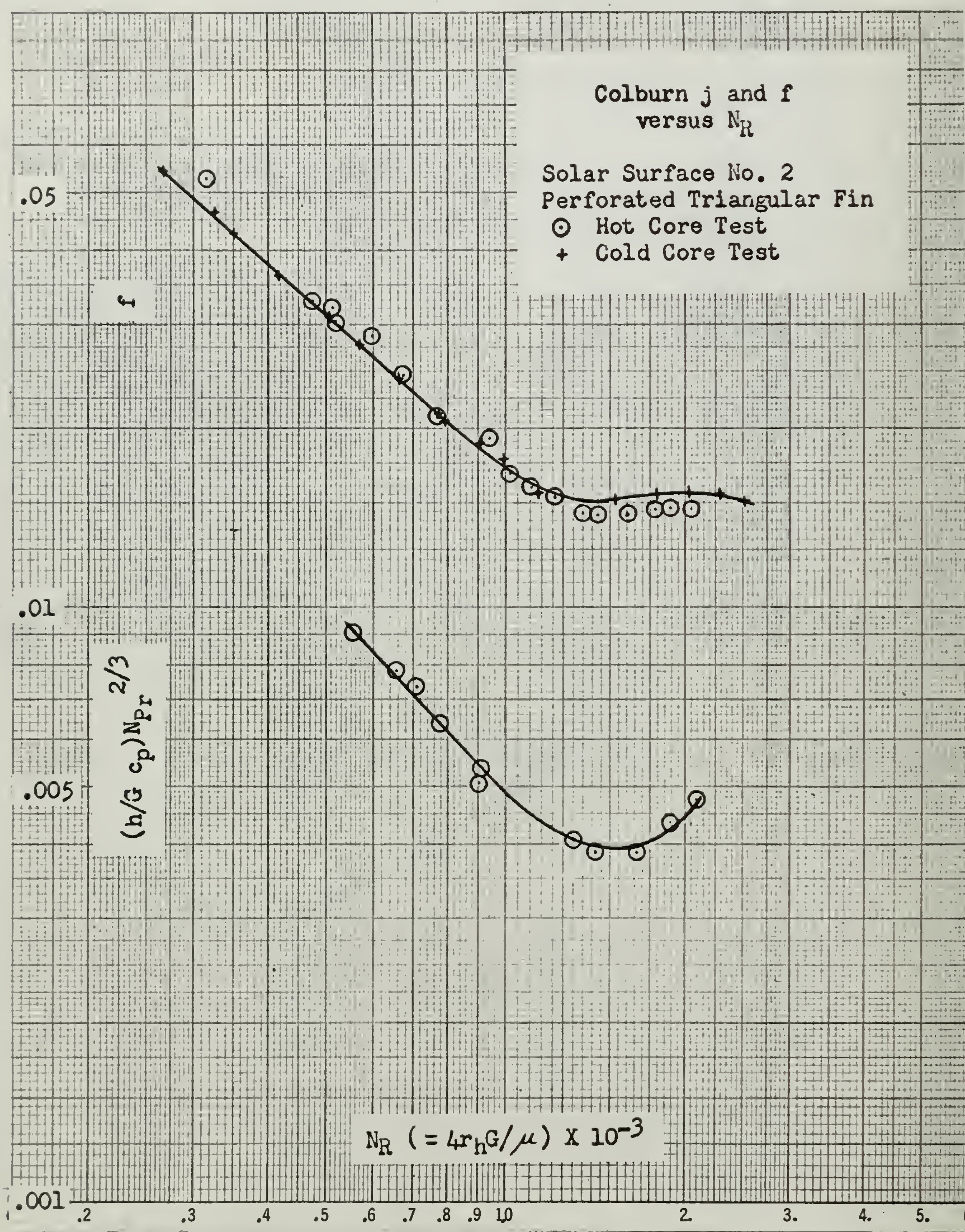


Figure 9. Plate-Fin Surface Heat Transfer and Friction Data, Solar Surface No. 2

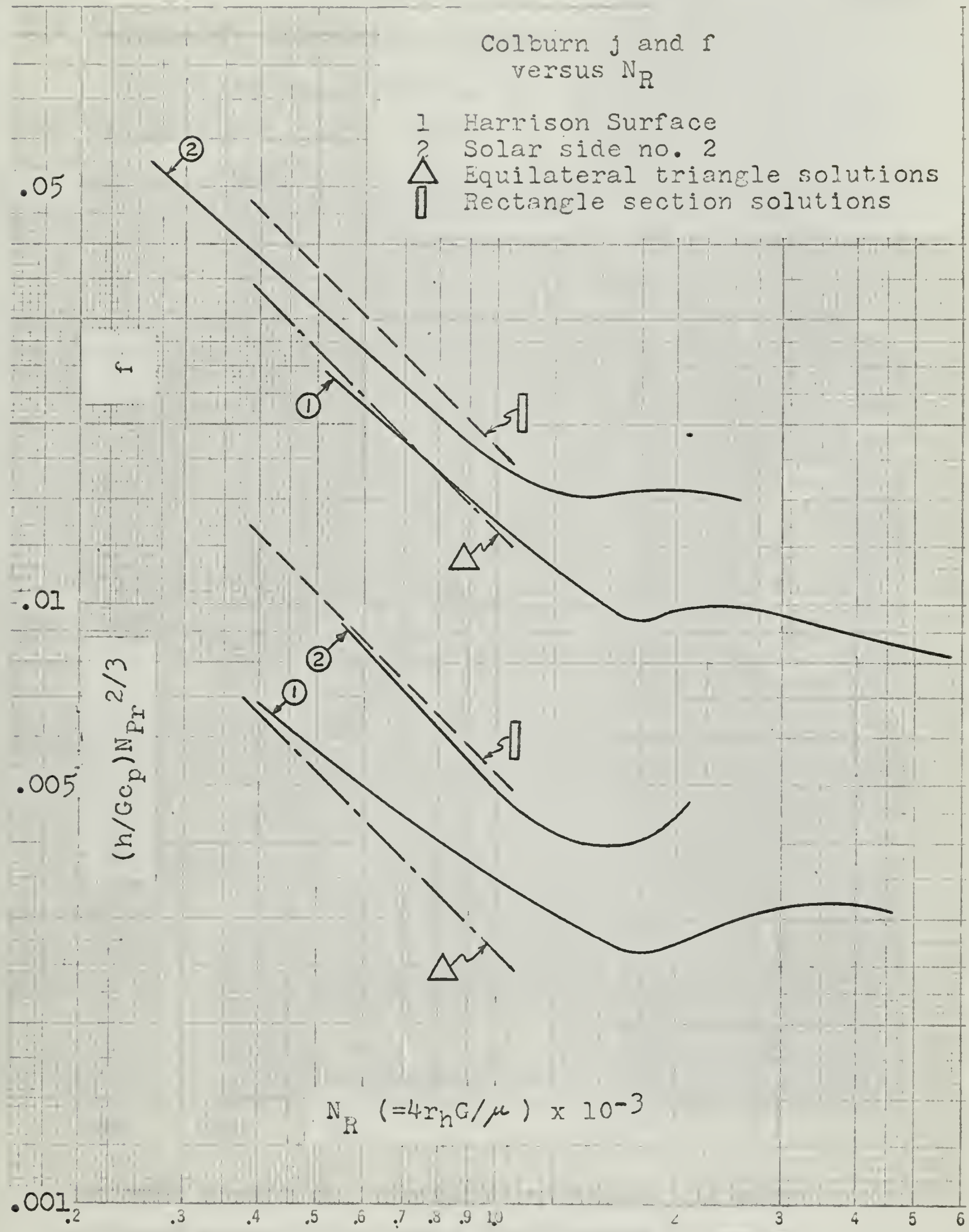


Figure 10. Summary and Comparison Curves

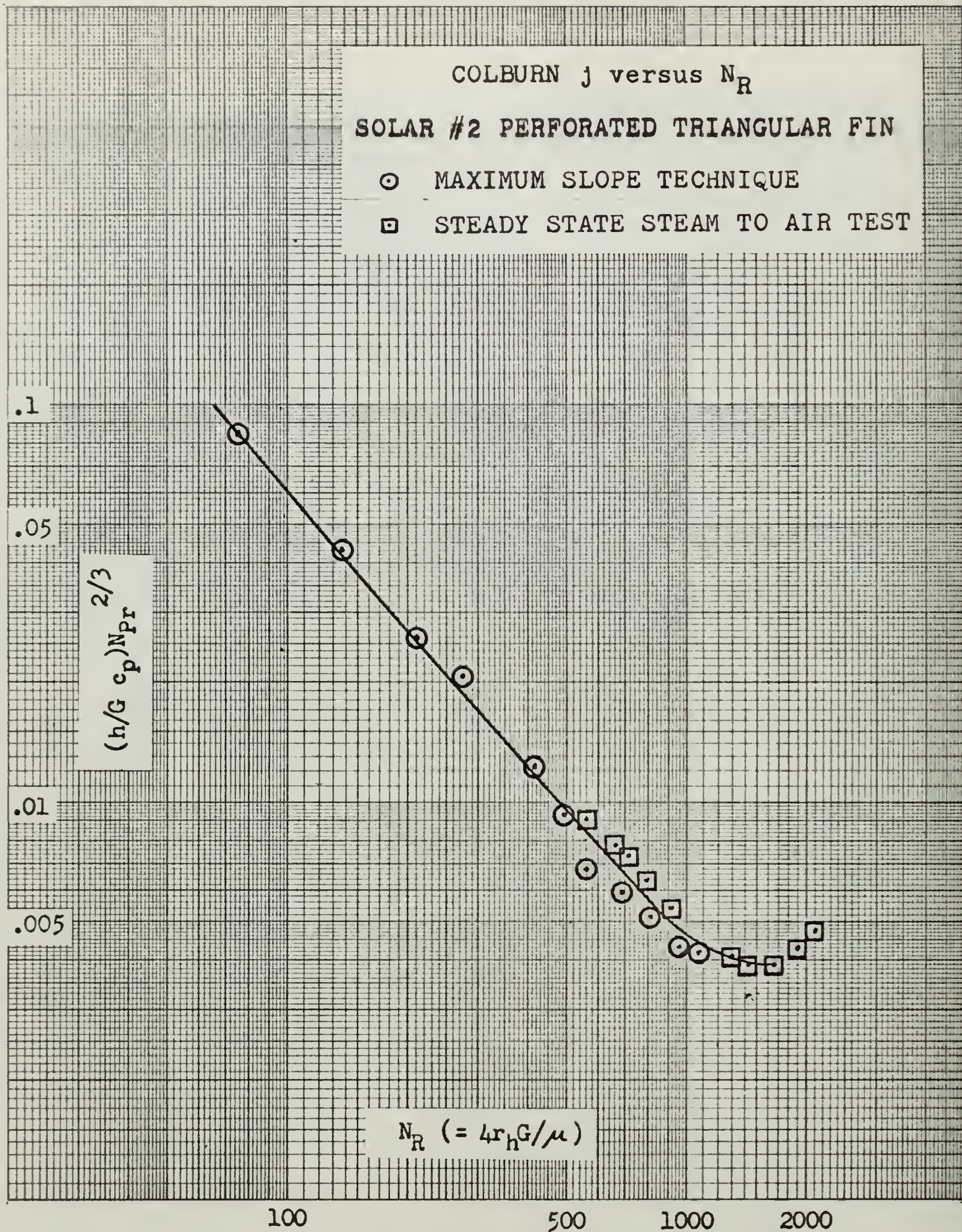
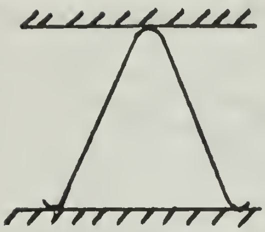
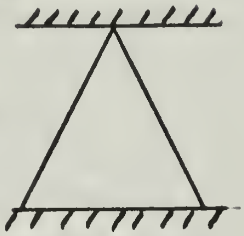
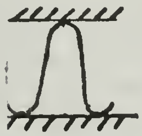
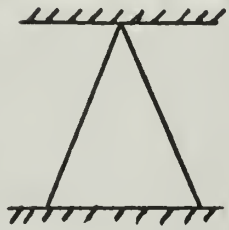


Figure 11. Comparison of Colburn j by Maximum Slope and Steady State, Steam-to-Air Testing Techniques

	RELATIVE GEOMETRY		
Surface	Actual	Idealized Common r_h	Curve No.
Harrison Surface No. 1			1
Solar Surface No. 2			2

It is noted that the Solar Surface No. 1 had a considerably larger radius of curvature at the fin base than did the Harrison Surface No. 1.

Figure 12. Comparison of Flow Cross-Sections

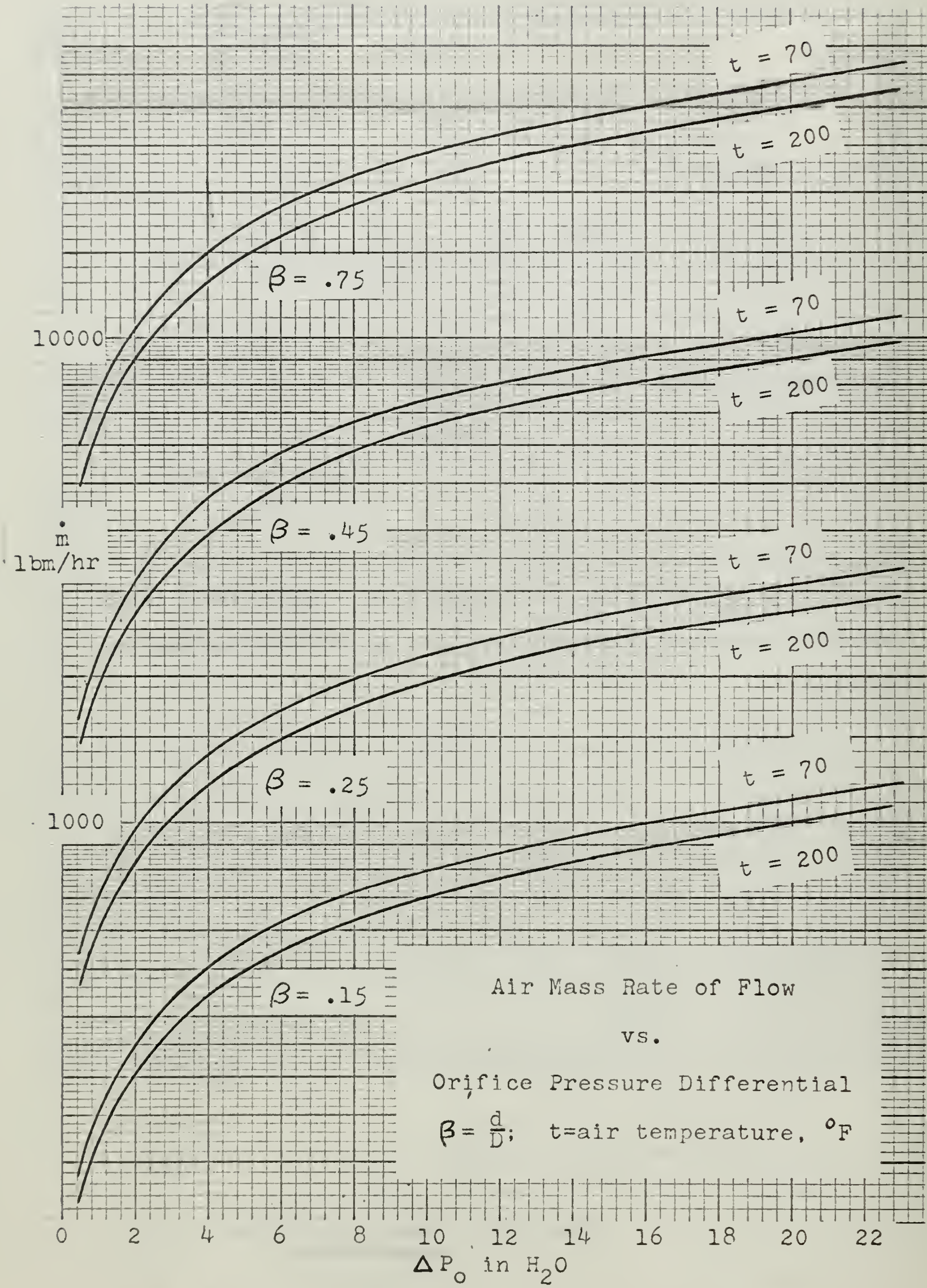


Figure 13. Mass Rate of Air Flow vs. Orifice Pressure Differential

30 APRIL 1966 SOLAR HOT CORE TEST RUN 32 - 37

RUN	-3.5 PS1	-3.5 PS2	DPS	PL	(5)	(6) DPC	(7)	(8) DPO	(9)	PO	PB	T_w	T_d	WC START	WC FINISH	WC	TIME	β_{air}	β_{atm}
32	16.00	15.00	14.50	.13	5.25			2.77		5.12	29.796	49.9	58.8	9.25	24.5	15.25	19.55	.25	.56
	16.10	15.00	14.51	.13	5.40			2.90		5.25		GRAINS = 39.5							
	16.12	15.08	14.50	.13	5.48			2.94		5.30									
	16.10	15.02	14.60	.13	5.48			2.95		5.30									
	12.58	11.52	14.53	.13	5.40			2.89		5.24									

23 APRIL 1966 SOLAR COLD CORE TEST RUNS 58-68

RUN	PL	(5)	(6) DPC	(7)	(8) DPO	(9)	PO		PB	T_w	T_d	β_{air}		TL	TO
58	.01	3.02			1.940		2.96		30.080	59.8	83.5	.25		1.17	1.20
	.01	3.03			1.950		2.97			GRAINS = 40.					
	.02	3.03			1.940		2.96								
	.017	3.027			1.943		2.963								

Figure 14. Sample Data Sheet for Hot and Cold Core Tests

APPENDIX I

TEST CORE GEOMETRICAL DATA, REDUCED LABORATORY DATA AND COORDINATES FOR THE BEST INTERPRETATION OF FRICTION AND HEAT TRANSFER SURFACE CHARACTERISTICS

This appendix furnishes the information needed for any reevaluation of test results.

Test Core Geometrical Data, Table 1

All test core dimensional data for use in the data reduction procedures, as specified by Kays [4] and Ward [12] are tabulated herein.

Reduced Laboratory Data, Table II

The reduced laboratory data for each test surface for both the hot core tests (Heat Transfer and Flow Friction Data) and cold core tests (Isothermal Friction Data) are tabulated.

Best Interpretation Surface Characteristics, Table III

Table III is a tabulation of the best interpretation of the reduced laboratory data, as taken from the curves for each surface in Figures 8 and 9.

TABLE I

TEST CORE GEOMETRICAL DATA

Harrison Radiator Core:

Surface No. 1:

Plate spacing -- .125 in

Plate metal thickness -- .012 in

Fin and plate material -- stainless steel of thermal conductivity, k_s = 10 BTU/(hr sq ft deg F/ft)

Frontal area -- .237 sq ft

Free-flow/frontal area -- .3353

Plate prime area -- 9.14 ft

Fin area -- 15.05 sq ft

Total heat transfer area -- 24.19 sq ft

Hydraulic diameter of flow passage -- $4r_h$ = .006724 ft

Air flow length -- .487 ft

Surface No. 2:

Plate spacing -- .125 in

Plate metal thickness -- .012 in

Fin metal thickness -- .012 in

Fin and plate material -- stainless steel of thermal conductivity, k_s = 10 BTU/(hr sq ft deg F/ft)

Frontal area -- .237 sq ft

Free-flow/frontal area -- .3353

Plate prime area -- 9.14 ft

Fin area -- 15.05 sq ft

Total heat transfer area -- 24.19 sq ft

Hydraulic diameter of flow passage -- $4r_h$ = .006724 ft

Air flow length -- .487 ft

Solar Core:

Surface No. 2, Perforated Fin:

Plate spacing -- .0985 in

Plate metal thickness -- .0047 in

Plate material -- solid nickel of thermal conductivity, k_s = 38.7 BTU/(hr sq ft deg F/ft)

Fin metal thickness -- .0047 in

Fin material -- perforated nickel (14 percent open area) of thermal conductivity, k_s = 38.7 BTU/(hr sq ft deg F/ft)

Frontal area -- .250 sq ft

Free-flow area -- .1056 sq ft

Free-flow/frontal area -- .422

Prime plate area -- 14.18 ft

Fin area -- 34.7 sq ft

Total heat transfer area -- 48.88 sq ft

Hydraulic diameter of flow passage -- $4r_h$ - .003854 ft

Air flow length -- .498 ft

Surface No. 1, Solid Fin:

Plate spacing -- .0985 in

Plate metal thickness -- .0047 in

Plate material -- solid nickel of thermal conductivity,

k_s = 38.7 BTU/(hr sq ft deg F/ft)

Fin metal thickness -- .0047 in

Fin material -- solid nickel of thermal conductivity,

k_s = 38.7 BTU/(hr sq ft deg F/ft)

Frontal area -- .250 sq ft

Free-flow area -- .1056 sq ft

Free-flow/frontal area -- .422

Prime plate area -- 14.18 sq ft

Fin area -- 40.4 sq ft

Total heat transfer area -- 54.58 sq ft

Hydraulic diameter of flow passage -- $4r_h$ = .003854 ft

Air flow length -- .498 ft

Table II Reduced Laboratory Data - Harrison Surface No. 1

HEAT TRANSFER AND FRICTION DATA

RUN	MDOT LB/HR	G LB/HR-FT2	HUMID LB/LB	T1 F	P1 " H2O	T2 F	DPC " H2O	TS F	H BTU/ HR-FT2-F	COLJ	F FACT	NR B	ERROR PERCENT
1	364.2	4361.1	.0086	57.0	410.0	197.0	.55	226.5	6.49	.00483	.02360	616.5	-7.9
2	513.4	6148.0	.0086	54.1	410.0	181.9	.88	226.2	7.11	.00376	.01837	879.3	-13.1
3	667.5	7994.3	.0080	54.0	410.1	171.2	1.22	226.4	7.77	.00316	.01446	1151.5	-10.4
4	908.2	10876.4	.0080	52.3	410.1	160.6	1.86	225.5	9.13	.00273	.01127	1579.6	5.7
5	1170.4	14016.6	.0080	61.4	410.5	164.2	2.83	224.7	11.98	.00278	.00977	2018.5	-2.3
6	1621.0	19413.7	.0080	58.8	410.4	168.7	5.00	224.4	18.46	.00309	.00973	2792.3	3.8
7	2072.7	24822.5	.0080	56.7	410.3	168.2	7.86	224.0	24.05	.00315	.00916	3576.3	2.4
8	240.8	2883.4	.0088	63.5	410.4	215.6	.34	229.4	6.10	.00686	.03340	401.2	-14.7
9	316.9	3795.1	.0088	63.9	410.4	206.3	.48	229.1	6.40	.00547	.02739	531.0	-11.5
10	565.5	6772.9	.0091	63.2	410.4	183.7	1.04	229.2	7.48	.00359	.01769	961.9	.1

Table II (continued) - Harrison Surface No. 1

HEAT TRANSFER AND FRICTION DATA

RUN	MDOT LB/HR	G LB/HR-FT2	HUMID LB/LB	T1 F	P1 ''H2O	T2 F	DPC ''H2O	TS F	H BTU/ HR-FT2-F	COLJ	F FACT	NR B	ERROR PERCENT
11	351.4	4208.2	.0091	66.2	410.4	202.5	.55	229.3	6.46	.00498	.02538	589.4	-7.8
12	866.0	10371.1	.0096	65.7	410.4	169.4	1.78	228.9	8.95	.00280	.01182	1484.2	.5
13	311.5	3730.3	.0106	67.7	409.0	206.3	.46	228.9	6.24	.00543	.02689	520.7	-11.8
14	369.1	4420.9	.0106	68.2	409.0	200.9	.58	229.7	6.49	.00477	.02394	619.0	-13.9
15	400.9	4801.7	.0120	65.7	408.9	197.4	.65	229.6	6.67	.00451	.02241	674.8	-14.2
16	498.5	5970.6	.0122	62.6	408.9	188.4	.86	229.5	7.15	.00389	.01877	845.7	-11.8
17	806.5	9658.5	.0120	63.5	408.7	170.6	1.62	229.4	8.59	.00289	.01254	1383.2	-2.9
18	865.3	10362.8	.0114	63.6	408.6	169.2	1.77	229.6	8.99	.00282	.01168	1485.2	-1.5
19	1120.8	13422.2	.0114	60.0	408.6	165.2	2.60	229.4	11.23	.00272	.00970	1933.4	1.7
20	381.4	4568.2	.0097	71.2	410.3	200.3	.61	229.5	6.58	.00467	.02345	638.6	-.6

Table II (continued) - Harrison Surface No. 1

HEAT TRANSFER AND FRICTION DATA

RUN	MDOT LB/HR	G LB/HR FT2	HUMID LB/LB	T1 F	P1 '' H2O	T2 F	DPC '' H2O	TS F	H BTU/ HR FT2 F	COLJ	F FACT	NR B	ERROR PERCENT
21	939.3	11248.7	.0097	67.3	410.3	167.4	2.13	229.6	9.24	.00267	.01212	1610.1	-1.4
22	1244.1	14899.0	.0090	64.1	410.1	169.5	3.06	229.9	12.98	.00283	.00900	2134.4	8.6
23	1364.4	16340.2	.0090	62.3	410.1	166.9	3.76	229.7	13.86	.00276	.01060	2347.5	4.4
24	2230.6	26714.3	.0109	62.1	409.8	172.2	9.06	229.6	25.42	.00309	.00895	3825.1	8.0
25	730.6	8750.0	.0094	68.8	409.5	176.2	1.39	228.7	8.33	.00309	.01318	1244.2	1.3
27	347.8	4165.7	.0067	56.0	410.8	200.5	.53	229.1	6.37	.00497	.02507	587.9	-14.1
28	377.0	4514.4	.0066	59.1	410.9	197.8	.59	228.9	6.51	.00469	.02343	637.0	-17.5
29	491.3	5884.2	.0064	55.5	410.8	186.8	.82	229.2	7.06	.00390	.01871	838.1	-10.1
30	767.8	9194.6	.0066	50.9	410.8	168.6	1.48	228.9	8.48	.00300	.01299	1329.5	.1
31	831.4	9956.5	.0091	65.1	409.8	172.2	1.66	229.7	8.96	.00292	.01194	1422.9	.7
32	841.8	10081.7	.0050	50.0	410.8	164.9	1.68	228.7	8.86	.00286	.01209	1462.2	-1.4

Table II (continued) - Harrison Surface No. 1

HEAT TRANSFER AND FRICTION DATA

RUN	MDOT	G	HUMID	T1	P1	T2	DPC	TS	H BTU/	COLJ	F FACT	NR B	ERROR
	LB/HR	LB/HR-FT2	LB/LB	F	" H2O	F	" H2O	F	HR-FT2-F				PERCENT
33	894.1	10707.8	.0091	67.3	409.6	168.2	2.00	229.5	8.91	.00270	.01274	1532.0	-6.0
34	2775.2	33236.1	.0109	64.5	409.4	170.6	13.57	229.0	30.89	.00302	.00846	4756.5	8.0
35	236.9	2837.3	.0088	68.2	409.1	215.4	.33	229.0	5.95	.00681	.03400	393.7	-.2
36	408.0	4886.4	.0088	59.9	409.1	195.7	.68	229.1	6.75	.00448	.02289	690.1	-2.5
37	407.7	4882.7	.0088	57.0	409.1	195.1	.67	229.1	6.74	.00448	.02242	691.1	14.1
38	501.9	6010.4	.0088	56.4	409.1	186.4	.87	229.2	7.15	.00386	.01884	855.8	.0
39	1122.1	13438.5	.0083	54.1	408.9	162.9	2.55	229.1	11.23	.00272	.00950	1946.3	3.7

Table II (continued) - Harrison Surface No. 1

ISOTHERMAL FRICTION DATA

RUN	MDOT	G	HUMID	T1	P1	DPC	F FACT	NR
	LB/HR	LB/HR-FT2	LB/LB	F	''H2O	''H2O		
1	5887.1	70504.3	.0000	66.3	406.2	41.20	.00585	10833.4
2	4883.4	58484.3	.0000	65.0	406.7	27.33	.00583	9004.3
3	4122.1	49366.3	.0000	65.0	407.0	19.16	.00586	7600.5
4	3525.2	42218.3	.0000	66.3	407.1	14.13	.00602	6487.1
5	2422.3	29010.2	.0000	65.9	407.3	8.80	.00893	4460.5
6	2116.3	25345.5	.0000	66.8	407.4	6.89	.00926	3891.9
7	1720.2	20601.5	.0000	68.6	407.5	4.72	.00972	3155.2
8	1428.1	17102.9	.0000	68.6	407.5	3.39	.01026	2619.4
9	1205.0	14431.2	.0000	68.1	407.5	2.47	.01059	2211.6
10	994.5	11910.3	.0000	68.1	407.6	1.98	.01131	1825.3
11	822.6	9851.9	.0000	65.9	407.6	1.30	.01074	1514.8
12	578.3	6925.5	.0000	56.4	407.8	.75	.01354	1080.0
13	509.1	6097.5	.0000	57.3	407.8	.66	.01594	949.6
14	367.7	4403.9	.0000	57.8	407.8	.42	.02040	685.4

Table II (continued) - Harrison Surface No. 1

ISOTHERMAL FRICTION DATA

RUN	MDOT	G	HUMID	T1	P1	DPC	F FACT	NR
	LB/HR	LB/HR-FT2	LB/LB	F	''H2O	''H2O		
15	294.7	3529.2	.0000	63.2	411.0	.30	.02334	544.8
16	366.0	4382.8	.0000	62.6	411.0	.40	.01945	677.2
17	414.4	4962.4	.0000	62.7	411.0	.47	.01741	766.6
18	435.8	5218.9	.0000	65.0	410.9	.50	.01663	803.5
19	512.8	6140.8	.0000	64.5	410.9	.62	.01431	946.1
20	571.7	6846.6	.0000	65.4	410.8	.73	.01331	1053.4
21	255.3	3057.3	.0000	64.5	410.8	1.00	.11733	471.0
22	762.2	9127.7	.0000	66.0	410.8	1.10	.01064	1403.3
23	882.2	10565.0	.0000	65.9	410.8	1.38	.00974	1624.4
24	1272.0	15233.1	.0000	68.6	410.7	2.67	.01028	2333.0
25	1904.8	22812.1	.0000	67.0	410.5	5.53	.00927	3501.8

Table II (continued) - Harrison Surface No. 1

ISOTHERMAL FRICTION DATA

RUN	MDOT	G	HUMID	T1	P1	DPC	F FACT	NR
	LB/HR	LB/HR-FT2	LB/LB	F	" H2O	" H2O		
26	2163.1	25905.4	.0000	68.1	410.5	6.97	.00895	3970.1
27	2781.2	33308.2	.0000	66.3	410.3	11.10	.00846	5118.0
28	3084.7	36942.4	.0000	67.2	410.2	13.46	.00824	5669.0
29	690.7	8271.8	.0000	60.7	410.7	.93	.01117	1281.6
30	1014.1	12144.4	.0000	68.6	410.7	1.78	.00929	1860.0

Table II Reduced Laboratory Data - Solar Surface No. 2

HEAT TRANSFER AND FRICTION DATA

RUN	MDOT LB/HR	G LB/HR-FT2	HUMID LB/LB	T1 F	P1 '' H2O	T2 F	DPC '' H2O	TS F	H BTU/ HR-FT2-F	COLJ	F FACT	NR B	ERROR PERCENT
11	1073.2	10162.6	.0056	56.9	405.4	219.4	5.40	229.0	15.86	.00507	.02517	811.9	-9.6
12	598.9	5671.7	.0087	66.8	405.4	229.3	2.60	229.3	25.88	.01480	.03785	447.6	-2.2
13	719.3	6811.4	.0057	68.6	398.2	228.2	3.23	228.6	21.85	.01041	.03212	537.3	-.8
14	943.5	8934.8	.0059	66.8	407.7	225.1	4.10	229.4	17.56	.00638	.02441	706.9	4.8
15	1088.2	10304.9	.0063	65.0	407.7	221.7	4.93	229.4	17.08	.00538	.02210	817.9	5.1
18	1428.7	13529.2	.0060	72.6	409.5	218.8	6.59	228.6	20.37	.00489	.01687	1070.8	20.8
23	1547.3	14652.4	.0081	66.8	408.8	213.2	7.30	229.1	18.48	.00409	.01601	1167.9	11.9
24	1683.7	15944.4	.0080	66.3	408.7	211.4	8.29	229.5	19.08	.00389	.01532	1272.7	13.0
25	1973.2	18685.5	.0081	68.6	408.6	210.7	10.04	228.8	22.33	.C0388	.01325	1490.0	16.8
26	2245.3	21262.5	.0080	67.7	408.5	214.3	14.17	228.8	28.32	.00432	.01438	1692.7	11.0
27	2495.7	23633.5	.0081	65.0	408.3	217.0	17.89	228.9	34.75	.00477	.01454	1881.4	6.0
29	782.3	7408.6	.0074	60.5	410.1	226.6	3.31	228.6	17.84	.00782	.02887	587.9	3.4
30	844.3	7995.3	.0069	60.9	409.9	226.0	3.61	228.6	18.15	.00737	.02699	634.5	2.7
31	666.0	6307.2	.0069	64.1	407.5	227.5	2.75	228.4	17.66	.00909	.03296	499.1	-4.7
40	2629.0	24895.4	.0063	67.2	408.3	211.1	19.85	226.3	32.67	.00426	.01467	1986.3	41.9

Table II (continued) - Solar Surface No. 2

ISOTHERMAL FRICTION DATA

RUN	MDOT	G	HUMID	T1	P1	DPC	F FACT	NR
	LB/HR	LB/HR-FT2	LB/LB	F	'' H2O	'' H2O		
50	972.6	9210.5	.0057	84.9	409.4	3.03	.02065	789.7
51	1402.5	13280.8	.0057	84.9	408.6	4.84	.01556	1138.7
52	1877.4	17778.1	.0081	81.4	409.3	8.50	.01523	1531.9
53	2185.6	20697.1	.0081	77.9	409.2	11.78	.01559	1792.5
54	2480.7	23491.6	.0059	77.9	409.1	15.21	.01556	2034.5
55	2797.5	26491.7	.0060	76.6	409.0	19.20	.01536	2298.7
56	3071.7	29088.1	.0060	77.0	408.9	23.05	.01518	2522.3
59	324.8	3076.2	.0060	76.1	409.3	.85	.05468	267.1
60	399.7	3784.6	.0060	75.9	409.3	1.09	.04606	328.7
61	427.8	4050.9	.0057	74.3	409.3	1.16	.04283	352.6
62	506.4	4795.1	.0057	73.5	409.3	1.38	.03638	417.9
63	614.7	5821.4	.0061	72.6	409.3	1.72	.03051	508.0
64	687.6	6511.3	.0057	71.7	409.2	1.96	.02777	569.0
65	801.8	7592.5	.0060	71.7	409.2	2.35	.02430	663.5
66	938.1	8883.5	.0058	70.3	409.2	2.83	.02126	777.8
67	1095.4	10372.9	.0057	69.9	409.2	3.47	.01902	908.8

TABLE III

SUMMARY OF BASIC HEAT TRANSFER AND FLOW FRICTION CHARACTERISTICS

j and f versus N_R from smoothed curves

HARRISON AIR SIDE			SOLAR SIDE NO. 2		
N_R	j	f	N_R	j	f
10000			10000		
9000			9000		
8000			8000		
7000			7000		
6000			6000		
5000		.00845	5000		
4000	.00320	.00890	4000		
3000	.00314	.00955	3000		
2000	.00273	.00980	2000	.00444	.0156
1500	.00277	.0100	1500	.00394	.0152
1200	.00311	.0118	1200	.00425	.0156
1000	.00346	.0136	1000	.00487	.0171
800	.00400	.0166	800	.00618	.0205
600	.00490	.0214	600	.00840	.0265
500	.00567	.0250	500		.0313
400	.00670		400		.0378
300			300		.0488
200			200		

APPENDIX II

OPERATING PROCEDURE

The operating procedure about to be presented has been based on many hours of actual operating experience aimed at reducing the warm-up, testing, and shut-down time to a minimum without subjecting the test rig to severe thermal stresses and sacrificing any accuracy due to steady state equilibrium not having been established.

It is recommended that prior to lighting off any equipment of this test facility, the literature listed below should be read carefully, especially the operating procedure and the equipment's capabilities and limitations. A copy of all literature is filed by Item Number in Building 500 with Mr. Joe Beck.

Item Number	Equipment	Literature Title
79	Spencer Blower	Spencer Instructions for Handling, Installing and Adjusting Spencer Equipment.
93	Centrifix Separators	Centrific Engineering Manual for Accurate Selection of High Efficiency Purifiers.
98	Leslie Reducing Valves	Instructions for Pressure Reducing Valves -- Small Flow "ATMO" Pressure Reducing Valves and Air Loaders. Instructions for Pressure Reducing Valves -- Installation, Operation, and Maintenance, GP Type Regulators.

Item Number	Equipment	Literature Title
100	Honeywell Recorder	Instruction Manual, ElectroniK 16, Multipoint Strip Chart Recorder.

General. The initial procedures are the same for both the heat transfer and friction factor tests (hot core tests), in which the steam system is energized to heat the core and for the isothermal friction factor tests (cold core tests), in which the core is not heated.

Remove the door and energize the Honeywell "ElectroniK 16" Multipoint Strip Chart Recorder and allow it to warm-up about two hours prior to calibration. To calibrate the recorder, set the print mechanism to "Hold On", indicating point number 1 (or any of the other points not connected to a thermocouple). To select a point, pull the instrument out about six inches, and on the left side (facing the recorder) is the "Select-O-Point" mechanism (a round disk with 24 numbers and "captive buttons"). To indicate number 1, pull button number 1 out. If all the other buttons are in, the recorder will cycle to number 1. To energize the amplifier, turn the chart drive mechanism to "LO". Place a Rubicon Potentiometer (a D.C. millivolt source) in front of the recorder and connect the long leads on the back of the recorder to the potentiometer connecting (+) terminals together. The calibrated accuracy of the recorder for its whole span is $\pm$ .25% of span, if the

recorder is calibrated to print the exact values at 20% and 80% of scale. Set two millivolts on the potentiometer and adjust the "zero" on the recorder to indicate 2. Set eight millivolts on the potentiometer and adjust the "span" until 8 is indicated. Repeat the "zero" and "span" adjustments until the instrument is calibrated. Intermediate voltages (i.e., 5 mv) should be applied, approaching from above and below to check the deadband range of the recorder (0.1% of full scale span) in order to insure that the "gain" (sensitivity) value is proper. Complete procedures are explained in the SERVICE section of the instruction manual. This calibration is needed because the chart paper can swell or shrink as much as a degree Fahrenheit or more, depending on the atmospheric conditions. Periodic maintenance, as specified in the instruction manual, should be performed to insure reliable operation.

Set the reference thermocouple on the back side in an "ice bath".

Zero all manometers with the isolation valves in the open position. To zero the three mercury manometers, the plugs in the top of the four water pots should be removed. Frequently, after a hot core test run, water will condense in some of the pressure lines and seal off the passage to the water pots. Steam then condenses in the top of the water pots,

creating a partial vacuum, and indicating an erroneous zero reading. The first two mercury manometers must be zeroed at about three and a half inches due to the column of water from the water pots to manometers. Whenever refilling the water pots to the manometers, bleed slowly and long, because these lines are very susceptible to the formation of air pockets.

The approximate range of testing of all heat exchanger cores is between a Reynolds number of 500 to 10,000. This range will vary slightly depending on the characteristics of the cores. To assure an even distribution of data points, calculate the mass rate of flow for these two Reynolds numbers and plot them on log-log paper. Divide the line between them into the desired number of points. Using Figure 13, pick off the approximate differential air pressure for each point and the desired air orifice plate needed.

Whenever installing an air or steam orifice plate, there is a certain specific position for each to obtain accurate metering. The air orifice plate has a "V" at the bottom and a rectangular lug at the top with a scribe mark and orifice diameter size stamped on it. The "V" fits over a brass dowl at the bottom of the orifice flanges and the scribe mark matches with another scribe on the downstream orifice flange. When installing the steam orifice plate, the steam flanges must

be centered vertically and horizontally with respect to each other and the steam orifice plate centered also. The writing on the rectangular lug should face upstream and the drain hole should be placed in the lowest position.

Hot core tests. It takes approximately one and a half hours to achieve steady state conditions for the first run. Each run takes between 20-40 minutes (20 minutes for the high air flow rates and 40 minutes for the low flow rates, so that an accurate condensate flow rate can be measured). Between runs it takes an additional 10 to 15 minutes to reestablish steady state conditions. The pressure manometers rapidly adjust to the new conditions, but the air temperature, especially the air orifice temperature, takes considerably longer.

Crack and slowly open the main steam gate valve over the boiler and open any intermediate valves that may be closed between the main steam valve and the globe valve at the entrance to the test rig.

After zeroing all manometers, close all isolation valves.

Energize the H.P. compressor and bleed the reservoir tank of any moisture. Close the valve that isolates the leak in the H.P. line. Bleed the two Leslie ATMO pressure regulators by opening the cock at the bottom of the regulators (approximately two minutes). Set the "six psig reducer" (the top ATMO regulator) at about six psig and "30 psig reducer" at about 30 psig.

Close the "desuperheater water" valve. Turn on the tap water supply for the condensate cooler and desuperheater.

Open the two Leslie steam reducer by-pass valves. Crack the steam entrance globe valve to allow the steam system to warm up slowly. When warm-up is completed, close the by-pass valves and fully open the steam valve. Now accurately set the inlet steam header pressure at six psig using the "PS1" mercury manometer (about 3.5 + 12.3 = 15.8 in Hg).

Insure that the blower discharge gate is closed and the OFF-ON switch beside the blower is in the ON position. Check to see that all air system manometers are closed. Throw the safety switch, above the started box, to the ON position. Throw the starter lever to the START position until the blower is up to speed and then throw the lever hard to the RUN position so that the electromagnet holds it.

Open the isolation switches to the 30-inch DPO (differential orifice pressure) and 60-inch PO (orifice pressure) manometers (numbers 9 and 10). Open the blower discharge gate as necessary, to increase the air flow and control the air flow with the sliding plate valve in front of the blower. An extra 60-inch manometer has been connected in parallel with the 30-inch DPO manometer to facilitate setting the desired orifice pressure differential.

Testing of the cores is done with the inlet steam state to the core at six psig with five to 10 degrees superheat. With the atmospheric pressure about 30 inches Hg, the absolute steam pressure to the core is about 40.3 inches Hg corresponding to a saturation steam temperature of 230 degrees F. Five to ten degrees superheat is 4.88 to 5.01 millivolts on the recorder. On the recorder "Select-O-Print" mechanism, pull out only bottons 2 and 3 (steam core inlet and outlet temperatures, respectively). Turn chart speed to LO, the Select-O-Print selector switch to "Select-O-Print" and then observe the steam temperatures. If the inlet steam temperature is too high, slowly reduce the steam pressure at the first pressure reducer and add "desuperheater water" as necessary until the desired temperature is reached. Then pull out bottons 2 through 17 for each test run.

For all tests the excess steam or "blow" steam should be at least five times the condensate flow-rate. For this, two steam orifice plates and a gate valve downstream of the steam orifice have been provided. Usually the gate valve is left fully open, but on very high heat transfer cores, low Reynolds number runs can only be achieved by cutting down the volume of steam.

The data is recorded for each run in the following order and to the indicated number of digits:

Item No.	Manometer Number	Designation	Description
1	1	PS1	Top steam header pressure (0.01 in Hg)
2	2	PS2	Bottom steam header pressure (0.01 in Hg)
3	3	DPS	Steam orifice pressure differential (0.01 in Hg)
4	4	P1	Core upstream pressure (vacuum) (0.01 in H_2O)
5	5,6,7	DPC	Core pressure differential (0.001 in H_2O for $\Delta P_c < 3"$, and 0.01 in H_2O for $\Delta P_c > 3"$)
6	8,9	DPO	Air orifice pressure differential (0.001 in H_2O for $\Delta P_o < 3"$ and 0.01 in H_2O for $\Delta P_o > 3"$)
7	10	PO	Air orifice upstream pressure (vacuum) (0.01 in H_2O)
8		PB	Atmospheric pressure (0.001 in Hg)
9		T_w	Atmospheric wet bulb temperature (0.1 deg F)
10		T_d	Atmospheric dry bulb temperature (0.1 deg F)
11		WC	Condensate weight (0.1 lbf)
12		TC	Condensate collection time (0.1 sec)
	Thermo-couple No.		
13	5,6,7	T1	Core upstream air temperature (0.01 millivolts)
14	17	TO	Orifice air temperature (0.01 millivolts)
15	8-16	T2	Core downstream air temperature (0.01 millivolts)

Item No.	Thermo- couple No.	Desig- nation	Description
16	4	TSO	Steam orifice temperature (0.01 millivolts)
17	3		Bottom steam header temperature (0.01 millivolts)
18	2	TS1	Top steam header temperature (0.01 millivolts)
19		IB	Air orifice size (nominal)
20		IBS	Blow steam orifice size (nominal)

Items 1 through 10 are recorded at approximately four equally spaced time intervals and averaged on the data sheet, making sure to subtract 3.5 (or zero level) from PS1 and PS2. Items 8 through 12, 19 and 20 are recorded once a run. Items 13 through 16, and 18 are averaged on the chart paper in millivolts.

Items 9 and 10 are measured with a sling-psychrometer and the weight of water vapor in one pound of dry air (in grains) is determined from a psychrometric chart.

A sample data sheet and sample run is shown in Figure 14.

Cold core tests. Once steady state is achieved, runs can be completed in ten-minute intervals.

This test is performed with the downstream air thermocouples removed. The data is recorded for each run in the following order and to the indicated number of digits:

Item No.	Manometer Number	Designation	Description
1	4	Pl	Core upstream pressure (vacuum) (0.01 in H_2O)
2	5,6,7	DPC	Core pressure differential (0.001 in H_2O for $\Delta P_c < 3"$, and 0.01 in H_2O for $\Delta P_c > 3"$)
3	8,9	DPO	Air orifice pressure differential (0.001 in H_2O for $\Delta P_o < 3"$ and 0.01 in H_2O for $\Delta P_o > 3"$)
4	10	PO	Air orifice upstream pressure (vacuum) (0.01 in H_2O)
5		PB	Atmospheric pressure (0.001 in Hg)
6		T_w	Atmospheric wet bulb temperature (0.1 deg F)
7		T_d	Atmospheric dry bulb temperature (0.1 deg F)
	Thermocouple No.		
8		Tl	Core upstream air temperature (0.01 millivolts)
9	17	TO	Orifice air temperature (0.01 millivolts)

Items 1 through 4 are recorded three times during each run and averaged on the data sheet. Items 5 through 7 are recorded once each run. Items 6 and 7 are measured with a sling psychrometer and the weight of water vapor in one pound of dry air (in grains) is determined from a psychrometer chart.

For items 8 and 9, the recorder chart paper can be stopped after each run and these values averaged while steady

state is being reestablished for the next run.

A sample data sheet and sample run is shown in Figure 14.

APPENDIX III

DIGITAL COMPUTER PROGRAM FOR DATA REDUCTION

This program, called SSHEAT, was written to reduce the raw data from the testing facility for both the hot core and cold core test. The program prints out the results in the same standard form used by W. M. Kays and A. L. London for both heat transfer and flow friction data, and for isothermal data. The program was written in FORTRAN 60 and a print out of it is shown on pages 98 through 105. A sample of the raw imput data for the hot core test is on pages 104-105. The output results for the hot and cold core test are on pages 66 through 74. The program glossary defines the input variables and those not mentioned are internal variables employed to define groupings for ease in programming.

The inputs to the program are the core parameters, identifying title, program indices, and core test raw data.

The core parameters are constant for each core and are read in on the first three cards, starting from AC and ending with ATA, and using floating point numbers. A standard heading is provided for the hot or cold core tests, but an additional identifying title for the print out is provided on the next input card in the alphabetic format, which can specify the date, the core, hot or cold core test, the specific runs

made and/or any other identifying information. The next (fifth) data card contains three items: NOR, IHORC, and ISQTRIN. NOR represents the number of runs and is used as a counter. NOR should be the highest numbered run and that particular run data should be placed last. IHORC is a program index which specifies whether all the data will be reduced as a hot or cold core test. The index is:

	IHORC
Hot core test	1
Cold core test	2

The next index is ISQTRIN for specifying the various fin geometries and is:

	ISQTRIN
Square fin	1
Triangular fin	2
Louvered or off-set (N_R = infinity)	3

NOR, IHORC, and ISQTRIN are fixed point numbers and must be right-adjusted in their specific fields. The next cards are the raw data cards for each run; three per run are required. The first of these three cards contains NR, IB, and IBS. NR stands for the particular run number and is self-explanatory. IB and IBS are indices for fluid metering orifices and are:

Air orifice		Air orifice index
Beta	d(in.)	IB
.15	2.081	1
.25	3.468	2
.45	6.244	3
.75	10.406	4

Steam orifice		Steam orifice index
BS	d_s(in.)	IBS
.560	0.700	1
.712	0.890	2

NR, IB, and IBS are fixed point numbers and must be right-adjusted in their specific fields. The second and third data cards for each run contains sixteen floating point items and the format varies for hot and cold core tests. All 16 items are needed for hot core tests and only eight for each cold core test, so that last data card for each cold core test is a blank card which the computer uses to zero all unused items.

The dimensions for the input data are specified in the program glossary and are the same as the recorded data.

PROGRAM GLOSSARY

Program Symbol	Nomenclature
AA	A_a = air side total heat transfer area (with perforations), sq ft
AC	A_c = exchanger air side minimum free flow area, sq ft
ALFA	σ = ratio of free flow area to frontal area
AFA	A_{fa} = fin area on air side, sq ft
AFR	A_{fr} = air side frontal area, sq ft
AFS	A_{fs} = fin area on steam side, sq ft
AKB	k_a = air thermal conductivity evaluated at bulk temperature
AS	A_s = steam side total transfer area, sq ft
ATA	A_{ta} = air side total heat transfer area (without perforations), sq ft
AWA	A_{wa} = prime plate transfer area on air side, sq ft
AWS	A_{ws} = prime plate transfer area on steam side, sq ft
BETA	β = ratio of air metering orifice diameter to duct diameter
BS	β_s = ratio of steam metering orifice diameter to duct diameter
C	C = coefficient of discharge of air orifice

Program Symbol	Nomenclature
CC	C_c = jet contraction ratio
CO	C_o = initial value of the coefficient of discharge for calculation purposes, air side
COLJ	j = Colburn j-factor
COS	C_{os} = initial value of the coefficient of discharge for calculation purposes, steam side
CP	c_p = specific heat of air at constant pressure
CRKD	K_{d_c} = velocity distribution coefficient for circular tubes
CS	C_s = coefficient of discharge of steam orifice
DC	ΔC = iteration interval in coefficient of discharge of air orifice
DCS	ΔC_s = iteration interval in coefficient of discharge of steam orifice
DD	D_d = hydraulic diameter of duct downstream of core, in
DL	l_d = length of duct downstream from core to pressure tap, in
DPC	ΔP_c = core differential pressure, in H_2O
DPO	ΔP_o = air orifice differential pressure, in H_2O

Program Symbol	Nomenclature
DPS	ΔP_s = steam orifice differential pressure, in H_2O
DSO	d_s = diameter of steam orifice, in
ERROR	Error = $(q_{air} - q_{stm})/q_{air} \times 100$
FA	F_A = thermal expansion factor for the fluid metering orifice, air side
FAS	F_{As} = same as FA, for steam side
FB	F = velocity of approach factor for the fluid metering orifice, air side
FBS	F_s = same as FB, for steam side
FC	f = Fanning friction factor in test core
FD	f_d = friction factor in duct immediately downstream of test core
FLA	l_a = fin length on air side
FLL	L = fin length flow direction
FLS	l_s = fin length on steam side
FKA	k_{sfa} = thermal conductivity of fin on air side
FKS	k_{sfs} = thermal conductivity of fin on steam side
FM	fm = friction factor of a passage in the test core
G	G = mass velocity, lbm/(hr sq ft)
GR	GR = grains of water vapor/lbm dry air
H	H = humidity ratio, lbm water vapor/lbm dry air

Program Symbol	Nomenclature
HA	h_a = thermal convection heat transfer on air side
HC	h_c = enthalpy of condensate leaving test core
HS	h_s = thermal convection heat transfer on steam side
HS1	h_{s1} = enthalpy of steam in top steam header
HS18	h_{s1} = enthalpy of steam in top steam header
H2	h_{s2} = enthalpy of steam in bottom steam header
IB	Index for air orifice diameter
IBS	Index for steam orifice diameter
IHORC	Index for hot or cold core test
ISQTRIN	Index for fin geometry
NOR	number of runs
NR	run rumber
OD	d = diameter of fluid metering orifice
P1	P_1 = pressure upstream of core, air side
P2	P_2 = pressure downstream of core, air side
PB	P_b = atmospheric pressure
PO	P_o = pressure at steam orifice
PR23	${N_{Pr}}^{2/3}$ = Prandtl number to 2/3 power
PS1	P_{s1} = pressure upstream of core, steam side
PS2	P_{s2} = pressure downstream of core, steam side

Program Symbol	Nomenclature
PSO	P_{os} = pressure at steam orifice
QAIR	q_{air} = heat transfer rate, air side
QSTM	q_{steam} = heat transfer rate, steam side
REB	N_R = Reynolds number
REO	N_R = Reynolds number at air orifice
RESO	N_R = Reynolds number at steam side
RH	r_h = hydraulic radius
RO1	ρ_1 = density of air entering core
RO2	ρ_2 = density of air leaving core
ROAVE	ρ_{avg} = density of air, average value
ROM	ρ_m = mean density of air in test core for hot core test
ROO	ρ_o = density of air metering orifice
SDOT	$\dot{s}$ = mass rate of steam flow
ST	N_{St} = Stanton number
SQKD	K_{d_s} = velocity distribution coefficient for square fins
T1	t_1 = inlet air temperature
T2	t_2 = outlet air temperature
TB	t_b = bulk air temperature
TC	time of condensate collection
TFA	δ_a = fin thickness, air side

Program Symbol	Nomenclature
TFS	δ_s = fin thickness, steam side
TO	t_o = temperature at air metering orifice
TS	t_s = saturated steam temperature corresponding to average steam pressure in core
TS1	t_{s1} = steam temperature at inlet to core
TS2	t_{s2} = steam temperature at outlet to core
TSO	t_{so} = steam temperature at metering orifice
TRKD	K_{d_t} = velocity distribution coefficient for triangular fins
TW	l_w = wall thickness between steam and air side of test core
U	U = unit overall thermal conductance
UB	μ = dynamic viscosity evaluated at bulk temperature
UC	μ = dynamic viscosity in core for cold core test
UO	μ_o = dynamic viscosity at air orifice
USO	μ_{so} = dynamic viscosity at steam orifice
VS	v_s = specific volume
WC	weight of condensate collected
WCH	$\dot{w}_c$ = mass rate of condensate from test core
WDOT	$\dot{m}$ = mass rate of air flow

Program Symbol	Nomenclature
WK	k_{sw} = thermal conductivity of wall between steam and air
X	x = ratio of pressure differential across orifice to upstream pressure, air side
XC	X_c = humidity correction to the specific heat of air
XKD	K_d = velocity distribution coefficient, general
XM	X_m = humidity correction to the density of air
XMA	m_a = $\sqrt{2h/k_{sf}\delta}$, air side
XMS	m_s = $\sqrt{2h/k_{sf}\delta}$, steam side
XNTU	N_{tu} = number of heat transfer units
XS	x_s = ratio of pressure differential across orifice to upstream pressure, steam side
Y	Y = net expansion factor for a square-edged orifice, air side
YKE	K_e = expansion coefficient
YNFA	η_{fa} = fin temperature effectiveness, air side
YNFS	η_{fs} = fin temperature effectiveness, steam side
YNO	η_o = total surface temperature effectiveness, air side
YNS	η_s = total surface temperature effectiveness, steam side

Program Symbol	Nomenclature
YS	Y_s = net expansion for a square-edged metering orifice, steam side
ZKC	K_c = contraction coefficient

```
..JOB0508F, RIDDELL     SSHEAT NO. 4
      PROGRAM SSHEAT4
      DIMENSION TITLE(10)
C  CAN READ IN TWO CARDS OF TITLE WITH PRINT OUT OF SAME IF DIMENSION OF
C  TITLE IS CHANGED TO TITLE(20)
      READ 1,AC,AFR,AA,AWA,AFA,AS,AWS,AFS
      READ 1,FLL,TFS,TFA,TW,FLA,FLS,FKA,FKS
      READ 1,WK,FD,DL,DD,ATA
    1 FORMAT(8F10.0)
      READ 10,TITLE
   10 FORMAT (10A8)
      READ 2,NOR,IHORC,ISQTRIN
    2 FORMAT(3I10)
      PRINT 811,TITLE
  811 FORMAT (1H1,1X,10A8///)
      GO TO (3,5),IHORC
    3 PRINT 4
    4 FORMAT(    25X,32H HEAT TRANSFER AND FRICTION DATA,///
     1106H RUN    MDOT       G      HUMID    T1     P1      T2     DPC    TS
     2H BTU/  COLJ    F FACT     NR      ERROR          /
     3 50H                                        ,,           ,,/
     4107H     LB/HR  LB/HR-FT2 LB/LB     F      H20     F      H20     F    H
     5R-FT2-F                          PERCENT          //)
      GO TO 7
    5 PRINT 6
    6 FORMAT(    20X,25H ISOTHERMAL FRICTION DATA,///
     1 71H RUN    MDOT       G         HUMID     T1      P1          DPC     F FAC
     2T       NR/
     3 50H                                        ,,           ,,/
     4 53H      LB/HR  LB/HR-FT2   LB/LB      F      H20        H20//)
    7 READ 8,NR,IB,IBS
    8 FORMAT (3I10)
C  READ IN TEMPERATURES AS MILLIVOLTS AND PROGRAM CONVERTS TO
C  DEGREES FAHRENHEIT.  TEMPERATURE SYMBOLS REMAIN THE SAME.
      GO TO (81,82),IHORC
   81 READ 9,GRAINS,T1,TO,T2,TSO,TS1,PB,PS1
```

```
      READ 9,PS2,DPS,P1,DPC,DPO,PO,WC,TC
    9 FORMAT (8F10.0)
      GO TO 83
   82 READ  9,P1,DPC,DPO,PO,PB,H,T1,TO
      READ  9,T2,TSO,TS1,PS1,PS2,DPS,WC,TC
   83 T1 =31.984974+46.819198*T1 -1.463541*(T1 )**2+.106778*(T1 )**3-
     1  .005383*(T1 )**4
      TO =31.984974+46.819198*TO -1.463541*(TO )**2+.106778*(TO )**3-
     1  .005383*(TO )**4
      T2 =31.984974+46.819198*T2 -1.463541*(T2 )**2+.106778*(T2 )**3-
     1  .005383*(T2 )**4
      TSO=31.984974+46.819198*TSO-1.463541*(TSO)**2+.106778*(TSO)**3-
     1  .005383*(TSO)**4
      TS1=31.984974+46.819198*TS1-1.463541*(TS1)**2+.106778*(TS1)**3-
     1  .005383*(TS1)**4
      H=GRAINS*.00014288
      PS1A=.4892*(PB+PS1)
      PS2A=.4892*(PB+PS2)
      PSAVE=(PS1A+PS2A)/2.
      TS=220.+(PSAVE-17.1)*2.66
      TOR=TO+459.7
      TOK=TOR*5./9.
      UO=.003527*TOK**1.5/(TOK+110.4)
       GO TO (20,21,22,23 ),IB
   20 CO=.59446
      DC=.00945
      C1=.5975
      OD=2.081
      BETA=.15
      GO TO 25
   21 CO=.59483
      DC=.01037
      C1=.5966
      OD=3.468
      BETA=.25
      GO TO 25
```

```
22 CO=.59868
   DC=.01543
   C1=.6014
   OD=6.244
   BETA=.45
   GO TO 25
23 CO=.60480
   DC=.05448
   C1=.6128
   OD=10.406
   BETA=.75
25 C=C1
   B4=BETA**4
   POA=.4892*PB-PO*.03605
   X=(DPO*.03605)/POA
   Y=1.-(((.41+.35*B4)*X)/1.4)
   XM=(1.+H)/(1.+1.607*H)
   ROO=POA*144.*XM/(53.3*TOR)
   IF(TO-68.)12,12,11
11 FA=1.+(TO-68.)*1.85E-5
   GO TO 13
12 FA=1.
13 FB=1./(1.-B4)**.5
   DO 26 I=1,3
   WDOT=359.*C*FB*FA*Y*OD*OD*SQRTF(DPO*ROO)
   REO=15.27 *WDOT/(13.875*UO)
   C=CO+DC*SQRTF(10000./REO)
26 CONTINUE
   GO TO (29,40),IHORC
29 TB=(T1+T2)/2.
   TBK=(TB+459.7)*5./9.
   UB=.003527*TBK**1.5/(TBK+110.4)
   AKB=.001529*TBK**.5/(1.+245.4/(TBK*10.**(12./TBK)))
   RH=AC*FLL/ATA
   G=WDOT/AC
   REB=4.*RH*G/UB
```

```
      XC=(1.+1.915*H)/(1.+H)
      CP= XC*.241
      DENOM=TS-T2
C   THIS IS A CHECK POINT TO PREVENT TOO HIGH A T2 FROM STOPPING THE
C   CALCULATIONS OF THE REST OF THE RUNS.  A PRINT OUT WILL INDICATE
C   THIS ERROR.
      IF(.00001-DENOM)90,91,91
   91 PRINT 92,NR
   92 FORMAT(/3HRUN,I3,44H AIR TEMP AFTER CORE GREATER THAN STEAM TEMP/)
      GO TO 7
   90 CONTINUE
      XNTU =LOGF((TS-T1)/(TS-T2))
      U=CP*WDOT*XNTU/AA
      HS=2000.
      XMS=SQRTF(24.*HS/(FKS*TFS))
      YNFS=(TANH(XMS*FLS/12.))/(XMS*FLS/12.)
      YNS=(AWS+YNFS*AFS)/AS
      HA = U
      DO 30 I=1,3
      XMA=SQRTF(24.*HA/(FKA*TFA))
      YNFA=(TANH(XMA*FLA/12.))/(XMA*FLA/12.)
      YNO=(AWA+YNFA*AFA)/AA
   30 HA=1./(YNO*((1./U)-AA*(1./(YNS*AS*HS)+TW/(AWA*WK*12.))))
      PR23=(UB*CP/AKB)**.667
      ST=HA/(G*CP)
      COLJ= ST*PR23
   40 ALFA= AC/AFR
      P2=P1+DPC
      P1A=PB*13.61-P1
      P2A=PB*13.61-P2
      T1A= T1+459.7
      T2A= T2+459.7
      RO1=.097318*XM*P1A/T1A
      RO2=.097318*XM*P2A/T2A
      GO TO (41,42),IHORC
   41 ROM=.5*((T1A*RO1+T2A*RO2)/(TS+459.7-((T2A-T1A)/XNTU)))
```

```
      GO TO 43
   42 ROAVE = .097318*XM*(P1A+P2A)/(2.*T1A)
      ROM=ROAVE
      G=WDOT/AC
      RH=AC*FLL/ATA
      T1K=(T1+459.7)*5./9.
      UC=.003527*T1K**1.5/(T1K+110.4)
      REC=4.*RH*G/UC
      REB=REC
   43 GO TO (50,45,44),ISQTRIN
   44 XKD=1.
      GO TO 60
   45 IF(REB -2000.)46,47,47
   46 XKD=1.43
      GO TO 60
   47 FM=.049/(REB **.2)
      CRKD=1.09068*4.*FM+.05884*SQRTF(4.*FM)+1.
      XKD=1.+1.29*(CRKD-1.)
      GO TO 60
   50 IF(REB  -2000.)51,53,53
   51 XKD=1.39
      GO TO 60
   53 FM=.049/(REB **.2)
      CRKD=1.09068*4.*FM+.05884*SQRTF(4.*FM)+1.
      XKD=1.+1.17*(CRKD-1.)
   60 CC =.611+.045*ALFA +.344*(ALFA**5.7)
      ZKC=(1.-2.*CC+(CC*CC)*((2.*XKD)-1.))/(CC*CC)
      YKE=1.-2.*XKD*ALFA+ALFA**2.
      A=(ZKC-(1.+ALFA*ALFA))/RO1
      B=(YKE+1.+ALFA*ALFA*(1.+(4.*FD*DL/DD)))/RO2
      FC=(RH*ROM/FLL)*(((4.3255E+9)*DPC/(G*G))-A-B)
      GO TO (70,104),IHORC
   70 GO TO (75,76),IBS
   75 COS=.6085
      DCS=.0250
      BS=.56
```

```
      DSO=.70
      GO TO 77
   76 COS=.6090
      DCS=.0410
      DSO=.89
      BS=.71
   77 CS=.61
      BS4=BS**4.
      FBS=1./(1.-BS4)**.5
      FAS=1.003
      DPSO=.4532*DPS
      PSO=17.1+(TSO-220.)*.376
      XS=DPSO/PSO
      YS=1.-(.41+.35*BS4)*(XS/1.3)
      DPSW=DPSO/.03605
      VS=24.01-(TSO-218.)*.4
      USO=8.25E-6+(TSO-200.)*1.45E-8
      DO 80 IS=1,3
      SDOT=359.*CS*FBS*FAS*YS*DSO*DSO*SQRTF(DPSW/VS)
      RESO=.004244*SDOT/(USO*1.25)
      CS=COS+DCS*SQRTF(10000./RESO)
   80 CONTINUE
      HS18=1158.0+(TS1-230.)*.5
      HS1=HS18-(PS1A-18.)*.35
      HS2=1155.3+(PS2A-19.)*.934
      HC=193.4+(PS2A-19.)*2.64
      WCH=WC*60./TC
      QAIR=WDOT*CP*(T2-T1)
      QSTM=WCH*(HS1-HC)+SDOT*(HS1-HS2)
      ERROR=(QAIR-QSTM)*100./QAIR
      WCHDH=WCH*(HS1-HC)
      SDOTDH=SDOT*(HS1-HS2)
      STOC=SDOT/WCH
      ZJPERFC=COLJ/FC
   99 PRINT 100,NR,WDOT,G,H,T1,P1A,T2,DPC,TS,HA,COLJ,FC,     REB,ERROR
  100 FORMAT(I3,2X,F6.1,2X,F7.1,2X,F5.4,2X,F4.1,2X,F5.1,2X,F5.1,2X,F5.2,
```

```
   12X,F5.1,2X,F5.2,2X,F6.5,2X,F6.5,2X,        F7.1,2X,F5.1,//)
    GO TO 110
104 PRINT 105,NR,WDOT,G,H,T1,P1A ,DPC ,FC,REB
105 FORMAT(I3,3X,F6.1,3X,F7.1,3X,F5.4,3X,F4.1,3X,F5.1,3X,F6.2,3X,F6.5,
   1 3X,F8.1//)
110 IF(NR-NOR)7,112,112
112 CONTINUE
    END
    END
```

```
.1056      .25       48.88     14.18     34.7      .1056     14.18     40.4
.498       .0047     .0047     .0985     .0493     .0493     38.7      38.7
38.7       .0051     11.125    6.        54.58
SOLAR HOT CORE TESTS  RUNS 11-21
        21          1         3
        11          2         1
.00565     .54       3.95      4.47        4.26    4.96      29.796    12.58
11.52      14.53     .13       5.4       2.89       5.24     15.25     19.545
        12          1         1
.00873     .76       3.73      4.73      4.27      4.92       29.79    12.87
11.82      15.2      .08       2.605     6.86      2.53      9.        22.388
        13          1         1
.00572     .8        3.82      4.7       4.25      4.94      29.266    12.63
11.99      15.37     .10       3.23      10.2      3.14      12.5      26.84
        14          1         1
.00586     .76       3.425     4.62      4.25      4.95      29.962    12.59
11.92      15.27     .105      4.105     17.       4.        15.5      26.8
        15          1         1
.00635      .72      3.92      4.53      4.25      4.94      29.962    12.56
11.89      15.24     .12       4.93      23.6      4.8       17.       25.715
        18          2         2
.006       .89       3.99       4.453     4.37      4.96     30.096     12.09
10.85      12.36     .09       6.59      5.13      6.42      17.       25.92
        19          2         2
.00687      .93      3.9       4.31      4.32      4.93      30.096    11.89
10.63      12.13     .14       10.02     9.12      9.8       20.       23.13
```

	20	2	2				
.00687	.92	4.02	4.439	4.3	4.81	30.094	12.65
10.54	11.83	.255	18.26	16.9	17.98	25.	19.635
	21	2	2				
.00687	.93	4.05	4.478	4.3	4.8	30.092	12.55
10.32	11.67	.32	24.33	22.59	23.84	25.	17.183

APPENDIX IV

TABULATED RESULTS FROM EVALUATION OF INSTRUMENTATION

TABLE I

Tabulation of Results of Temperature Check
Comparing Thermocouple and Thermometer Readings

TC	mv	deg F
2	.86	71.26
3	.85	70.82
8	.84	70.36
9	.85	70.82
11	.85	70.82
12	.85	70.82
14	.86	71.26
T_{Hg}	---	71.15

TOTAL mv = 5.96

AVG. mv = .851

SCATTER = $\pm$ .01(.45 deg F)

AVG. mv = .851 = 70.84 deg F

T_{Hg} = 71.15

Difference = - .31 deg F

TABLE II

Calculations from the Steam
Saturation State Check

RUN 1 29 March 1966 HARRISON CORE

Measured values: $t_{s2} = 4.675$ mv

$P_{s2} = 9.70$ in Hg

$P_b = 30.125$ in Hg

Calculations: $t_{s2} = 4.675$ mv $= 227.2$ deg F

$$
\begin{array}{rcl}
P_{s2} & = & 9.70 \text{ in Hg} \\
\underline{+\ P_b} & = & \underline{30.125 \text{ in Hg}} \\
P_{s2A} & = & 39.825 \text{ in Hg}
\end{array}
$$

From Keenan and Keyes [7] the corresponding saturation temperature is 226.8 deg F

Comparison: temperature measurement is 0.4 deg F higher than pressure.

RUN 24 1 May 1966 SOLAR CORE

Measured values: $t_{s2} = 4.75$ mv

$P_{s2} = 12.00$ in Hg

$P_b = 30.040$ in Hg

Calculations: $t_{s2} = 4.75$ mv $= 230.0$ deg F

$$
\begin{array}{rcl}
P_{s2} & = & 12.00 \text{ in Hg} \\
\underline{+\ P_b} & = & \underline{30.040 \text{ in Hg}} \\
P_{s2A} & = & 42.040 \text{ in Hg}
\end{array}
$$

From Keenan and Keyes [7] the corresponding saturation temperature is 229.6 deg F

Comparison: temperature measurement is 0.4 deg F higher than the pressure.

APPENDIX V

COMPUTER PROGRAM FOR CONVERTING MILLIVOLTS TO DEGREES FAHRENHEIT

The results of this program were incorporated into the master program to reduce considerably the time required to process the raw data into a form acceptable for the computer program. Now the tedious job of interpreting in the tables of millivolts vs degrees fahrenheit has been eliminated with no significant loss in accuracy.

The problem was to fit a curve through five points, approximately equidistant apart on the millivolt scale and covering the range of temperature used. This was done by writing five, fourth order, simultaneous equations using the five chosen points as solutions to the five equations. The five points are:

^{o}F	Millivolts
77	.990
121	2.011
161	3.007
202	4.018
240	5.014

The five equations are:

$$77 = x_1 + x_2(.990) + x_3(.990)^2 + x_4(.990)^3 + x_5(.990)^4$$

$$121 = x_1 + x_2(2.011) + x_3(2.011)^2 + x_4(2.011)^3 + x_5(2.011)^4$$

$$162 = x_1 + x_2(3.007) + x_3(3.007)^2 + x_4(3.007)^3 + x_5(3.007)^4$$

$$202 = x_1 + x_2(4.018) + x_3(4.018)^2 + x_4(4.018)^3 + x_5(4.018)^4$$

$$240 = x_1 + x_2(5.014) + x_3(5.014)^2 + x_4(5.014)^3 + x_5(5.014)^4$$

The program for establishing the constants $(x, \ldots x_5)$ is:

```
..JOB0508F RIDDELL  CONVERSION OF MILLIVOLTS TO DEG FAHRENHEIT
      PROGRAM MILVOLT
C
C  THE A ARRAY IS THE ARRAY OF THE COFFICIENTS OF THE UNKNOWN
C  CONSTANTS,(X1------X5).
C
C  THE X ARRAY IS THE ARRAY OF THE UNKNOWN CONSTANTS.
C
C  THE C ARRAY IS THE ARRAY OF THE FIVE SOLUTIONS TO THE
C  SIMULTANEOUS EQUATIONS.
C
      DIMENSION A(100,6),X(100),C(5)
      C(1)=.990
      C(2)=2.011
      C(3)=3.007
      C(4)=4.018
      C(5)=5.014
      A(1,6)=-77.
      A(2,6)=-121.
      A(3,6)=-162.
      A(4,6)=-202.
      A(5,6)=-240.
      DO 1 I=1,5
      A(I,1)=1.
      A(I,2)=C(I)
      CIS=C(I)**2
      A(I,3)=CIS
      A(I,4)=CIS*C(I)
      A(I,5)=CIS*CIS
    1 CONTINUE
      PRINT 10,((A(I,J),J=1,6),I=1,100)
   10 FORMAT (6F10.3)
      CALL JORDAN2(A,5,X)
      PRINT 20,X
   20 FORMAT (5F15.6)
      END
      END
```

The constants, determined by PROGRAM MILVOLT, are:

$X_1 = +31.984974$

$X_2 = +46.819198$

$X_3 = -1.463541$

$X_4 = +.106778$

$X_5 = -.005383$

PROGRAM TEST was written to check out PROGRAM MILVOLT to determine if the constants were sufficiently accurate for other points in between the five chosen points. The following is PROGRAM TEST:

```
..JOB0508F RIDDELL  TEST OUT OF MILVOLT PROGRAM
      PROGRAM TEST
      PRINT 8
    8 FORMAT(1H1,//48H    TO           T1           T2           TS1          TSO    )
    9 READ 44,TO,T1,T2,TS1,TSO,NUMB
   44 FORMAT(5F10.0,I10)
      TO =31.984974+46.819198*TO -1.463541*(TO )**2+.106778*(TO )**3-
     1  .005383*(TO )**4
      T1 =31.984974+46.819198*T1 -1.463541*(T1 )**2+.106778*(T1 )**3-
     1  .005383*(T1 )**4
      T2 =31.984974+46.819198*T2 -1.463541*(T2 )**2+.106778*(T2 )**3-
     1  .005383*(T2 )**4
      TS1=31.984974+46.819198*TS1-1.463541*(TS1)**2+.106778*(TS1)**3-
     1  .005383*(TS1)**4
      TSO=31.984974+46.819198*TSO-1.463541*(TSO)**2+.106778*(TSO)**3-
     1  .005383*(TSO)**4
      PRINT 45,TO,T1,T2,TS1,TSO
   45 FORMAT(/2X,F7.2,2X,F7.2,2X,F7.2,2X,F7.2,2X,F7.2/)
      IF(NUMB-3)9,10,10
   10 END
      END

.990      2.011     3.007     4.018     5.014          1
.389      1.517     3.458     4.486     5.147          3
```

The check points selected and the computed values are compared below and are well within the accuracy needed.

MILLIVOLTS	TABLE VALUE($^{\circ}$F)	COMPUTED VALUE($^{\circ}$F)	ERROR
.389	50	49.98	.02
1.517	100	99.99	.01
3.458	180	180.03	.03
4.486	220	220.02	.02
5.147	245	244.97	.03
.990	77	77.00	.00
2.011	121	121.00	.00
3.007	161	162.00	.00
4.018	202	202.00	.00
5.014	240	240.00	.00

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Basic heat transfer and flow friction characteristics are presented for two different plate-fin compact heat exchanger surfaces employing the steady state, steam-to-air testing technique. One surface is a plain triangular fin of stainless steel and the other is a triangular fin fabricated from perforated nickel.

The experimental heat transfer characteristics of the perforated nickel fin obtained by the steady state steam-to-air testing technique, described herein, is compared with the results of an identical fin tested by the maximum slope (or transient test) technique.

Both surfaces tested compared favorably with their corresponding analytical solutions; and the comparison of the perforated fin by the two different test techniques was very good.

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